Note 2

Power (Horsepower and Accuracy)

Output in this review is quoted in Horsepower (HP) rather than converted to Kilowatts (even although the latter are a compliment to the man who conceived Horsepower!), and Torque is in Lb.Wt.Ft not Newton Metres (another compliment!). This is because it is believed that most readers, like the author, will be more at home with the historical units in a history.

The engine test methods can be considered as "SAE-standard", i.e.:-

- Unobstructed and unheated* inlet (no air cleaner);
- No dynamo or alternator;
- No engine fan unless air-cooled;
- No exhaust silencer - no Grand Prix engine has been silenced;
- Ignition and fuel/air mixture adjusted to optimum at each speed.

Although every care has been taken to obtain correct data, no distinction has been drawn generally in the text between British "Brake Horsepower" (BHP) and the 1.4% smaller Continental Horsepower (French "Cheval Vapeur" (CV); Italian "Cavallino Vapori" (CV); German "Pferdstarke" (PS)), because it is certain that the figures quoted by many makers over the years are nowhere near that accuracy. Apart from the variation in test-bed equipment and methods (steady settled conditions or "flash"), there are good reasons why this should be so:-

Racing engines are modified continuously and published specifications and powers may not always be consistent;
Some nominally-identical units are better than others for undiscovered differences;
Tests are brief because parts lives are short;
Few units are available;
And, to cap it all, the competition must be misled!

The broad sweep of power across the years is the effect at which the author has aimed.

There are particular cases where there is good evidence that quoted powers were as much as 10% optimistic and these are described in Note 5 (Delage) and Note 6 (Maserati)

On the other hand some makers have given figures in KW, CV or PS which can be trusted and a reliable conversion can be made to BHP and this has been done.

"Rated power"

A special point about power is that it has sometimes *not* been the "Natural Peak" value but some lower figure because engine speed has been limited mechanically or thermally, i.e. the engine has been "Rated" so as to achieve a desired life. Without a power curve for each engine – and these are fairly rare – it cannot be known where the given figure is "Rated".

---

*Cosworth found that the usual empirical correction used in testing:-

Power proportional to √[Absolute ambient Temperature]

was insufficiently accurate when looking for small power changes from modifications *versus* the range of temperatures possible in an ordinary working day (cold at the start, hot by midday).

Therefore they built a test cell with an air conditioning system for the inlet so as to be able to run at a constant temperature. The figure chosen was that expected at the circuit where the engines would be raced. A typical level was 25^{0}C (1114)

Note 5

Delage power 1925 – 1927

Most data sources for the Delage racing cars in 1925 and 1927, which have been selected here as CoY, give the following powers and crank speeds:-

			HP	@ RPM
Type 2LCV	2 Litre	1925	190	7,000
Original type 15-S-8	1.5 Litre	1926	160	7,500
Improved "	"	1927	170	8,000

These are the powers entered in Appendix 1 (sources quoted there) but there are good reasons for believing that they are exaggerated as sustainable figures. These reasons are given below, considering the 1927 1.5 L engine.

1. Pomeroy (4) made the point that the car was provided with a 5th "overdrive" gear ratio which reduced RPM by 19% compared with direct drive. This shows that the designer, Lory, did not expect the engine to sustain 8,000 RPM, only 6,500 RPM. The tachometer was coloured green from 6,000 to 7,500 RPM (39).
   It is not doubted that the engine *could* exceed 8,000 RPM. When Campbell bought two of these cars in 1928 his mechanic, Leo Villa, was told by the works that the maximum permitted speed was 8,400 (226). Later on, when Giulio Ramponi was testing the engine which he modified for Dick Seaman (see Item 4), it is said that he accidentally over-revved to 9,000 RPM for several minutes without mishap (39). This overspeed reliability without piston/valve collision was probably a large factor in the car's successes, although Item 5 below does suggest that high rpm could not be sustained for long without piston ring flutter fatigue.
2. The 1927 Delage claim can be compared with a contemporary test result on the FIAT type 406 2 x IL6 1.5 L, which was not a published claim with all the possible intent to mislead but works data only revealed in 1950(66):-

	1927	
	Delage Claimed	FIAT Works test
Compression Ratio (R)	6.5	7 from drawing
Inlet Valve Pressure (IVP)	1.51 ATA	1.85 ATA
HP	170	160
@ RPM	8,000	8,000

From a 3% lower power (by ASE, Air Standard Efficiency) and 18% less from IVP the Delage could have been expected to have rather less than 130 HP. It might have been rather more than this because its 55.8mm bore versus the FIAT's 50mm would have produced a higher Combustion Efficiency (EC) by having a lower (Surface Area)/(Chamber Volume) ratio. At the most this would be an improvement from 65% to 70%, so the Delage might have reached around 140 HP.

3. On the Brooklands Outer Circuit the 1926 Delage lapped at only 117.91 MPH (4), far below the speed expected from a general study of performance on this course (732) if its power had really been 160 HP. However, there must have been some problem such as driver inexperience of the bumpy, banked track, because a 1927 type in the hands of Lord Howe achieved 127.05 in 1931 (645). Even this speed, with a claimed 170 HP, is poor when compared to the performance in 1935 of Freddy Dixon in his naturally-aspirated 2 L Riley Special. This was a car of very similar size and shape to the Delage i.e. open-wheeled, low-slung and a narrow "2-seater" width. The Riley did have an elongated tail which presumably did add a trifle to the speed or Dixon would not have used it! Dixon's lap, actually made during a 500 mile race, was 134.4 MPH, 5.8% higher than the Delage on a power not exceeding 150 HP (141).

4. When Guilio Ramponi in 1936 developed the 1927 Delage engine for Dick Seaman to race in Voiturette events**, his changes and the results compared with the works specification as follows:-

	1927 Claimed	1936	
R	6.5	7.5*	New pistons (521) probably slipper-type with reduced friction.
IVP	1.51ATA	1.82ATA* (445), obtained by speeding-up the supercharger (659).	
Valve timing		"Modified"	
HP	170	185	Tested by Laystall (39, 521)
@ RPM	8,000	8,000	
		7,000 was considered the normal maximum in races (489).	

*These increases in R and IVP were probably made possible by increasing the alcohol %age in the fuel compared to the works Delage's Elcosina (which was 44%ethyl alcohol, 55% benzole, 1% castor oil), although the car was still able to race 200 miles nonstop.

The 185 HP tested by a respected company suggests that the original 1927 specification, with 5% and 17% less power from R and IVP respectively, would have yielded rather under 150 HP.

**A picture of the Seaman Delage is given in Note 46, which speculates that Enzo Ferrari may have been influenced in proposing the Tipo 158 to Alfa Romeo by the performance of this car in 1936. After the season he would have been able to obtain details of the modified Delage from Ramponi, who was an old colleague at Alfa.

5. In Note 13 Part II on Piston Ring Flutter it is shown that a plain iron ring fitted with normal loose radial clearances has a flutter limit of:-
(Ring Axial Width (w) x Max. Piston Deceleration (MPD)) = 4,000 mm.g.
The 1927 Delage had w = 1.5 mm (enlarged section from (4)) and consequently would have entered ring flutter at MPD = 2667 g. This corresponds to 7,000 RPM.
Therefore any higher RPM could only have been held briefly before flutter broke up the rings in fatigue, losing power because of blow-by and with crankcase oil blown overboard. This is really a clinching point.

Probable cause of exaggerated power

A 10% drop in power between a "flash" brake reading(i.e. one obtained by taking an engine to high RPM while still cold)and steady settled-temperature reading is quite possible. This is shown for Maserati engines in Note 6. The MG designer H.N.Charles commented that, on the highly-supercharged Q-type of 1934, even an additional 18% of power was available above the steady figure for a short period which was lost on throttling back to avoid detonation as the engine warmed up (331).

Deducting 10% from 170 would give 153 HP.

Conclusion

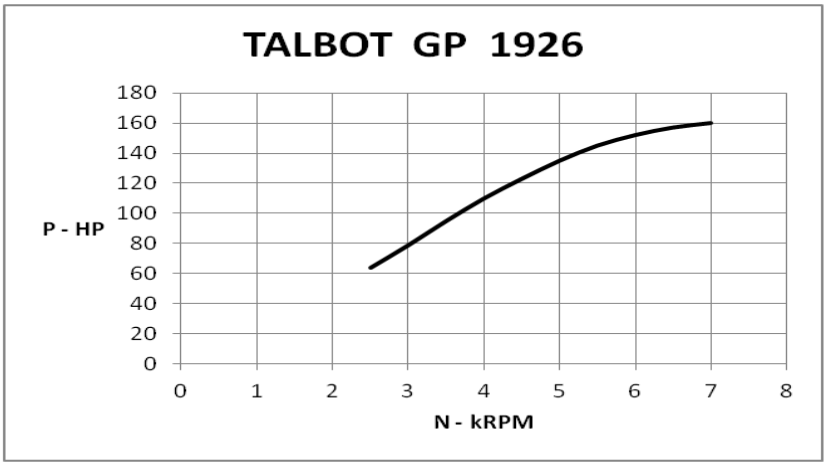
This Delage case has been treated at length because of the importance, not to say even worship, attached to the 1925 – 1927 designs.

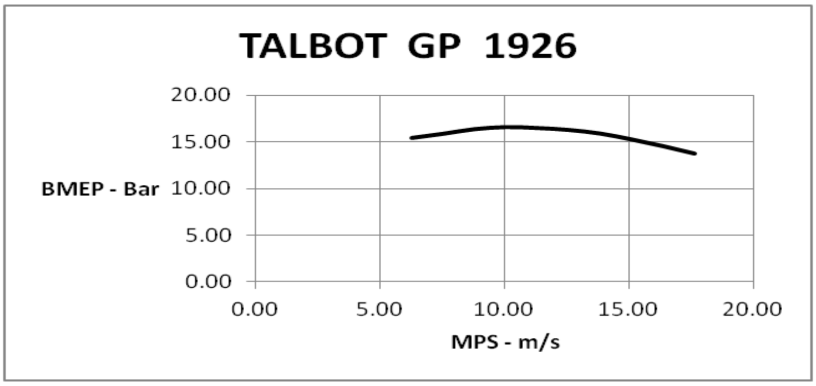
It is concluded that, to be comparable with other engines, the power of these Delage engines should be reduced by 10% relative to the usually published figures, at about 150 HP @ 7000 RPM for the 1927 15-S-8 and 170 HP @ 6,200 RPM for the 2LCV, because they were produced unsubstainably by "flash" testing,

P.S. A power curve for the 1926 Talbot IL8 1.5 L rival to the Delage is attached which shows 160 HP @ 7,000 RPM. This car was faster than the Delage in the only race in which they both competed, the 1926 Brooklands GP. Incurable front brake judder slowed the Talbots and the vibration probably caused the broken axle and two split supercharger casings from which the team of 3 cars retired.

POWER CURVES

Eg.	For comparison with Eg. 13			
DASO	12			
YEAR	1926			
Make	Talbot			
Model	GP			
Vcc	1488			
Ind. System	MSC			
Confign.	IL8			
Bmm	56			
Smm	75.5			
	N	P	MPS	BMEP
	kRPM	HP	m/s	Bar
	2.5	64	6.29	15.40
	3	79	7.55	15.84
	3.5	95	8.81	16.32
	4	110	10.07	16.54
	4.5	123	11.33	16.44
	5	135	12.58	16.24
	5.5	145	13.84	15.85
	6	152	15.10	15.23
	6.5	157	16.36	14.53
	7	160	17.62	13.75





Note 7

Power/Weight ratio, 1889 to 1998

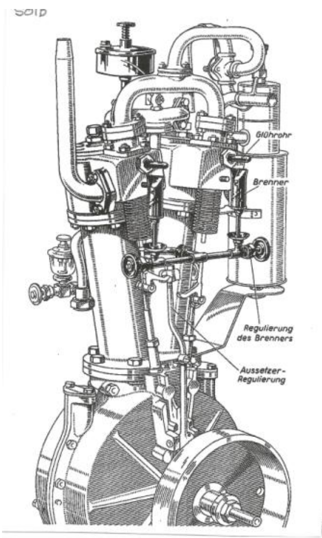
The development of the 4-stroke petrol-fuelled piston engine is even more remarkable in the century or so from 1889 when Daimler produced the 1st unit specifically for an automobile, as shown below compared with the Daimler-Benz-financed Ilmor FO110G racing engine of 1998:-

[*Italics* = approximate data]

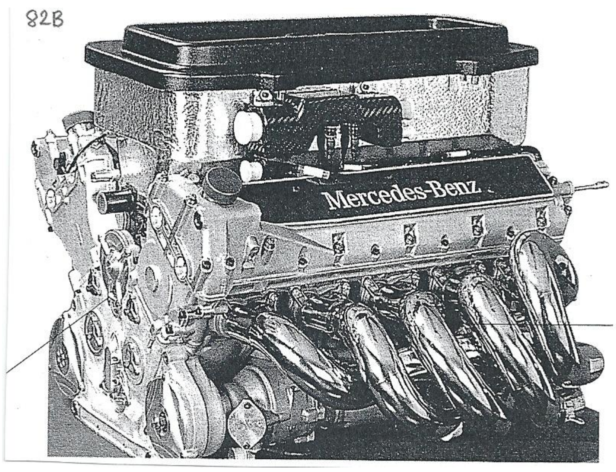
Date		1889	1998	
Data sources		(518,637)	(559)	
Vehicle		Stahlrad Wagen	McLaren MP4/13	
Engine Designer		Wilhelm Maybach	Mario Illien	
Configuration		16^0V2	72^0V10	
Bore/Stroke	mm	60/100 = 0.6	*93.5/43.67 = 2.14*	
Swept Volume	cc	566	2,998	
Power	HP	*1.5*	750	
Weight	kg	60	105	
Power/Weight	HP/kg	0.025	7.14	x 286!

While the 1998 engine was only intended for a life of about 3 hours (say, 400 km) before overhaul, it is doubtful if the 1889 design could have lasted much longer since its two narrow connecting-rods bore together on one very-skinny crankpin.

1889 Daimler



1998 Ilmor FO110G



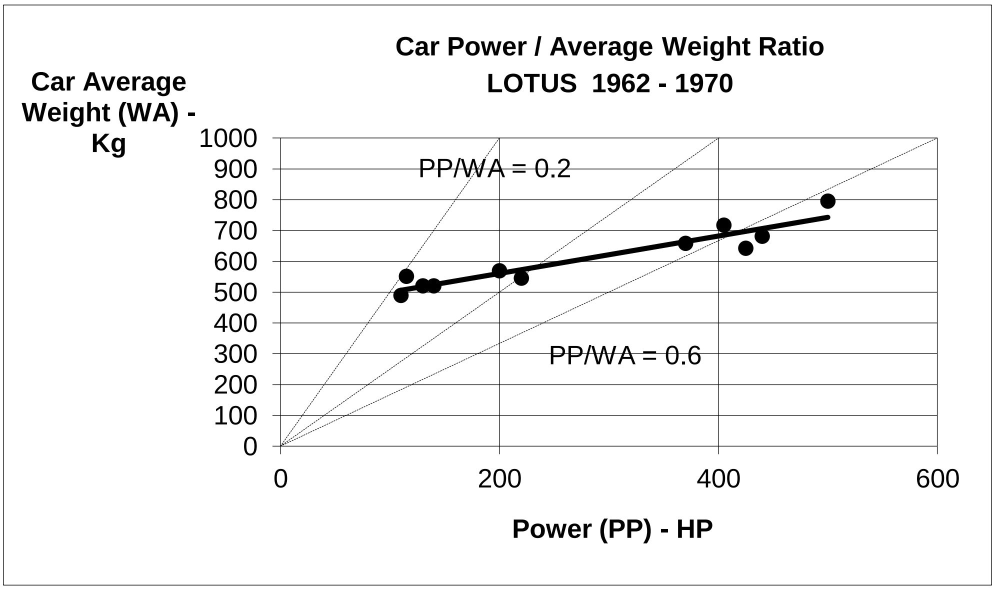
Note 8

Car Power/Average weight ratio

LOTUS 1962 – 1970 Technology:- Al-alloy “Monococque” Chassis

Type	Formula	Power PP - HP	Weights kg Unladen*	1/2 Fuel*	Driver*	Average (WA)
25	GP	200	451	42	76	569
27	FJ	110	400	*13*	76	489
29	Indy	370	513	69	76	658
32	F2	115	451	*24*	76	551
35	F2	130	420	*24*	76	520
38	Indy	500	612	107	76	795
43	GP	405	567	*74*	76	717
44	F2	140	420	*24*	76	520
48	F2	220	420	*49*	76	545
49	GP	425	500	66	76	642
72	GP	440	531	*74*	76	681

*It is *assumed* that Oil and Coolant are included, but effect would not be significant. Some fuel loads are approximate, shown in *Italics*; Driver is *assumed* as 12 stone.



As can be seen, Power/Weight ratio rose with installed power because a proportion of average car weight does not scale up with power. The same pattern was observable with the Grand Prix and Voiturette cars built by Mercedes-Benz, Alfa Romeo and Maserati in the 1938 - 1940 period, each pair to the same technology. For example the Mercedes-Benz cars compared as follows in PP/WA ratio (both 2-stage Supercharged)

1939 W154/M163 3L: 464 HP / 1145 kg = 0.4

1940 W165 1.5L: 274 HP / 902 kg = 0.3

Note 9

Exceptions to "Power 1st"

Certain individual races where a lower-powered car beat its rivals are evergreen in enthusiasts' minds but they needed "Super-Drivers"

- Tazio Nuvolari driving an Alfa Romeo in the 1935 German GP;

- Stirling Moss driving a Lotus in the 1961 Monaco and German GPs;

- Ayrton Senna driving a McLaren in the 1993 European (Donington) GP.

These successes sometimes were helped by wet roads.

The *major season-long exceptions* to the rule "Power 1st " are:-

(A).The 1959 -1960 Grand Prix Cars-of-the-Year (Egs. 38 & 39, Coventry Climax engines)

These were the fruits of the Cooper's (father Charles and son John) reintroduction to GP racing of the mid-mounted engine - but this time with a stable chassis - which gave frontal area and weight advantages more than offsetting the lower power. Since 1959 every classic Grand Prix has been won by this chassis concept (see Note 66) *bar one* (the 1960 Italian, not seriously contested against the front-engined Ferrari because the other significant teams boycotted the use of the Monza banked track variant).

(B).The 1966 – 1967 Grand Prix Cars-of-the-Year (Egs. 45 & 46, Repco engines)

The driver who was champion in 1959 - 1960, Jack Brabham, was the constructor of these cars bearing his name and, with characteristic shrewdness, he opted for a simple design of engine at the start of a new formula which gained victories while other more elaborate competing units were un-developed or not even ready. The Ferrari team which *should* have been ready and able in 1966 quarrelled with its 1964 Champion driver, John Surtees, and let him go after he won the 2nd race.

(C).The 1994 Drivers' Championship

This Championship result for Michael Schumacher driving a Benetton B194 was in inverse proportion to the installed power for the first three places:-

- 1st a V8 engine, the Ford Zetec R (Cosworth type EC, Eg. 77)
- 2nd a V10 engine, the Renault RS6 (Eg. 78) (driver Damon Hill) (the Williams FW16 won the Constructors' Championship)
- 3rd a V12 engine, the Ferrari type 043 (driver Gerhard Berger) (Ferrari 412T).

This result was despite the reintroduction of in-race refuelling which favoured the more powerful and thirstier engines.

However, the result was largely influenced by Schumacher being in the "Super-Driver" class, which enabled him to succeed even despite a disqualification and 2 race ban after ignoring a black flag and another disqualification for a worn skidblock. The death of Ayrton Senna at the third race, the other "Super-Driver" who was driving for the Williams-Renault team, also affected the result.

Note 10
POWER and FUEL

The basic 4-stroke internal-combustion piston-engine formula for power is:-

Equation 1
$$P = \frac{BMEP \times V \times N}{894{,}890}$$

where
P =Brake HorsePower (PP = Peak BHP)
BMEP = Brake Mean Effective Pressure; Bar (BMPP = BMEP at Peak Power);
(BMTP = BMEP at Peak Torque)
V = Swept Volume; cc
N = Crank Revolutions per Minute (RPM) (NP = RPM at Peak Power).

The basic expression for BMEP (following Ricardo, Ref. 242) is:-

Equation 2
$$BMEP = \left[\frac{10 \times DM \times (LCV + LH) \times RSV}{1 + SAFR}\right] \times MDR \times ASE \times EV \times EC \times EM;\ \text{Bar}$$

where
DM = Density of (air + fuel) mixture; kg/m^3 .
LCV = Lower Calorific Value of fuel; MJ/kg
LH = Latent Heat of fuel at constant volume; MJ/kg
RSV = Ratio of Specific Volume increase, after/before combustion
SAFR = Stoichiometric Air/Fuel mass Ratio
MDR = Manifold Density Ratio when Pressure-Charged (PC) as a ratio of the value of DM when Normally-Aspirated (NA)
ASE = Air Standard Efficiency = $\left[1 - \frac{1}{R^{0.4}}\right]$
where R = Compression Ratio
EV = Volumetric Efficiency
EC = Combustion Efficiency
EM = Mechanical Efficiency,including both Friction and Pumping losses. These latter would include any net effect when Pressure-Charged, which reduces EM when Mechanically-Supercharged (MSC), and increases it when Turbocharged (TC).

Equation 2 is given as an *aide memoire* to the way that engines have been developed, but no details of efficiencies have been published for successful GP units (to the author's knowledge) - and, in most cases until recently, have probably never been measured! Ricardo's 1922 3L Vauxhall engine is the only known published example of a racing design analysed in such a way (Ref.242).

It was , of course, due to Ricardo's initiative that Tizard and Pye established in 1919 that the value of the bracketted term [ ], which is the "Ideal" MEP calculated before any efficiencies are applied, is constant to within 2% for all useable volatile liquid fuels (Refs.242,343,728). It is about **38 Bar at Standard Temperature and Pressure** (STP, i.e. 15C and 1.01325 Bar (14.696psi)). The significant differences found between fuels were:-
(1.) their resistance to knock in combustion and therefore in the Compression Ratio which could be used;
(2.) the greater evaporative cooling of alcohols which thereby improved Volumetric Efficiency.

As the alcohols (ethyl and methyl) were found by Ricardo to be highly knock-resistant as well as charge- and engine-coolers, he and Halford pioneered their use as a base for racing fuels in 1921 in a modified Triumph 500cc motor-cycle at Brooklands. Shortly after mechanical supercharging by Roots blower (RSC) was established for GP engines in 1923, alcohol-base fuels became the norm and they continued generally as such through the subsequent reversion to NA in 1952 until 1957. The parallel development of fuels in the commercial motor and aviation fields to higher knock-resistance (see Note 58) then led the companies which supplied racing fuels free-of charge for advertising purposes to press for the regulation use of 5 Star 102 Octane pump petrol "the same as you can buy at your local garage" in 1958 and thereafter. Actually AvGas 100/130 Grade was accepted 1958-1960, but 102 Research Octane Number (RON) petrol subsequently until 1983. Turbocharging (TC) had been introduced in 1977 and its special needs led the fuel suppliers to conclude that winning in itself was the major advertisement bonus and a competitive reversion occurred to special blends. These met the 102 RON test in a NA calibration engine but were much more knock-resistant than "Real Petrol" at high values of MDR in the racing units (Note 90). Post-1988, when PC was banned, the same circumvention of the spirit of the rules continued, e.g. fuel used in a 1992 Honda NA 3.5L engine produced 5% more power quite legally than "Real Petrol" to the same 102 RON (69). It seems that

this was also obtained by permitting higher compression pressure but possibly such fuels beat the "Tizard-Pye Law" by small amounts (535). Fuels were even optimised differently for Qualification over a few laps and for the race. However, as shown on TABLE 1, of "The Sporting Limits"", the rules were then tightened to exclude "Power-Boosting-Additives" meaning, in effect, "Anything not in pump petrol". APPENDIX 2 gives examples of fuels used in racing over the 1906-2000 period (these data, for general interest, cover a broader range than "GP Car-of the Year").

Returning to the formula for power, combining Equations 1. and 2. and inserting the "Tizard-Pye Law" gives, near enough:-

Equation 3 $$P = \frac{MDR \times V \times N \times ASE \times EV \times EC \times EM}{23{,}550} \quad \text{HP at STP}$$

on Hydrocarbon fuel.

It is usual to achieve this NA power with hydrocarbon fuel by running 20% richer than SAFR (SAFR/1.2 = 12.3 (242)), which increases flame speed (594) and overcomes the effect of chemical dissociation ("un-burning"). The Ricardo 1922 Vauxhall 3L TT engine power curve was run at that carburetter setting, after which, to obtain the lowest Specific Fuel Consumption (SFC), it was then re-tested with a 10% weak setting (SAFR/0.9 = 16.3).*

Tests show that the gain on using Methyl Alcohol fuel in a modern NA engine at SAFR, from the greater evaporative cooling, other things being equal, is 12% compared to petrol (Ref.55. Ricardo found it to be 8% in a 1921 engine, Ref. 242). It is also possible with alcohol to use a very rich mixture (down to SAFR/1.4) to gain up to a further 10% over petrol when NA (Ref.586). This is apart from gain in ASE by raising R. Details vary according to engine design.

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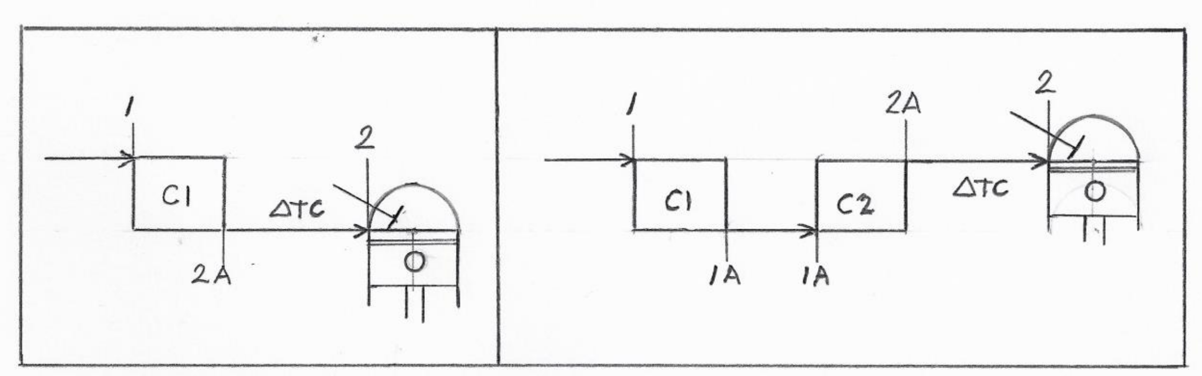
*

Unfortunately, Ricardo did not publish the SFC data for the 20% rich "Power" test, which restricts analysis. It has not been possible to obtain the missing data from the company on recent requests. There was a Peak Power drop of 6% from the 20% Rich Mixture test to the 10% Weak setting (242).

Note 10B

Estimation of Manifold Density Ratio (MDR) for Pressure-Charged (PC) engines

In this Note the flow stations through the Pressure-Charging system are identified as shown on the following diagrams:-



1-Stage Compression   2-Stage Compression

It is assumed that entry conditions are Standard ambient, i.e.

P1 = 1.0133 Bar (14.7 psi); T1 = 288^0K.

MDR, as a multiple of ambient air density, which is required in the basic Power equation (Eqn. 3 of Note 10) and which is taken to be 1 for Naturally- Aspirated (NA) engines, depends upon the absolute charge temperature at the inlet valve, T2, where:-

$$MDR = IVP/(T2/T1)$$

(T2/T1) depends upon:-

- The Efficiency of Compression (CE) to the pressure at the inlet valve (IVP = P2/P1);
- The post-Compression cooling (ΔTC) by fuel evaporation* or by intercooler. For the 1st method ΔTC depends on
  the Latent Heat of the fuel constituents;
  and the richness of the Air/Fuel Ratio (AFR).

- It is assumed in the calculations for simplicity that ΔTC occurs after the Compressor has added heat, although all Grand Prix CoY in the 1st PC Era 1924 – 1951 except the 1924 Alfa Romeo P2 and the 1935 Mercedes-Benz M25C had their fuel sprayed into the airstream by a carburetter upstream of the supercharger. In the 2nd PC Era 1983 – 1988 fuel was injected post-Compression.

The Temperature rise is given fundamentally by:-

$$\Delta T12A = T1 \times \left((IVP)^{0.286} - 1 \right) / CE$$

1st PC Era: 1924 – 1951

For the Roots-type mechanical superchargers used exclusively for Grand Prix CoY between 1924 and 1951, an empirical expression by Maleev (641) was used in this review which gives:-

$$\Delta T12A = T1 \times \left((IVP)^{0.5} - 1 \right)$$

This has implicit values of CE:- at IVP = 1.3, CE = 55%; at IVP = 2.1, CE = 53%.

An IVP of 2.1 is the highest figure for a Roots supercharger before the efficiency falls off drastically. Data derived from (468) shows that, at IVP 2.43 used in the 1938 Mercedes M154, CE = 35%. No other GP CoY operated in the region between 2.1 and 2.4 and it was realised by Mercedes in 1938 that higher than 2.1 the compression should be shared between 2 superchargers in series, each within the “normal” Roots efficiency range around 50%.

In this review for the calculation of MDR with 2-stage supercharging it was assumed that the 1st and 2nd stage pressure ratios were equal, i.e. at √(IVP).

Fuels

The fuels used for each CoY engine are given in Appendix 1, referenced to details in Appendix 2 Table.

Air/Fuel Ratio (AFR)

No data are available on the mixture strengths used in the engines considered in this review so a figure has been assumed according to their circumstances.

For 50% Petrol + 50% Benzole and for the widely-used “Elcosine” (53% Benzole + 44% Ethanol, etc,) over 1924 – 1934, a 20% richness for maximum power seemed reasonable. In 1934 Mercedes introduced ‘WW’ 86% Methanol fuel and they were concerned about its high consumption so the chemically-correct AFR ratio would have been appropriate, as it would be for the lower-alcohol mixture used in 1935 – 1937. The inefficient M154 of 1938 would have needed ‘WW’ at full power 44% richness to cool its supercharger outflow. The 2-stage more-efficiently-supercharged M163 would have needed a substantially less-rich ‘WW’ mixture to avoid cooling below ambient temperatures which would have caused icing-up in wet weather. This M163 had a full-throttle super-rich feature on its carburetter which sometimes jammed, including during the German GP which was run in cold and wet conditions that did upset the carburetion set up in practice.

The 2-stage supercharged post-WW2 Alfa Romeo of 1948 to 1951 was assumed to require steadily-enhanced richness with its 98% Methanol fuel as boost pressure was increased until reaching full 44% richness in 1951 when IVP = 3.9. This mixture would have been unable to deduct all the compression heating.

2nd PC Era: 1983 – 1988

In this 2nd PC Era, which was exclusively Turbo-Charged (TC) in Grand Prix CoY, fuel to 102 RON was the regulation, which the authorities intended to mean Petrol. The CoY PC engines actually ran on an artificial fuel which met the Octane limitation in the low-speed control engine but gave superior performance in the racing engine (see Note 90). The ΔTC available from evaporation was only 22^{0}C at the max.-power richness of 20%. Consequently most of the post-Compression cooling needed to restore charge density and avoid knocking was provided by intercoolers between Compressor and inlet valve. All GP CoY engines used air-to-air intercoolers. These were made large enough, even at IVP = 4 Bar, to restore T2 to 40^{0}C above T1 for maximum power with 84% Toluene + 16% Heptane fuel, egs the 1987 – 1988 Hondas (20). Therefore this ratio of (T1/T2) = 0.92 was used in the estimates of MDR for all TC GP CoY engines since all ran on high-Toluene fuel.

Note 11

Peak Power condition

From Eqn. 3 of Note 10:-

$$P = \frac{MDR \times V \times N \times ASE \times EV \times EC \times EM}{23,550} \quad \text{HP. on Petrol; at STP.}$$

For a given engine

$$P = \text{Constant} \times N \times (EV \times EC \times EM)$$

For convenience let Constant = K and (EC x EV x EM) = ECOM, so

$$P = K \times N \times ECOM$$

Take logs:-

$$\text{Log } P = \text{Log } K + \log N + \text{Log } ECOM$$

and differentiate, so that:-

$$\frac{dP}{P} = \frac{dN}{N} + \frac{dECOM}{ECOM}$$

At P = Peak Power, PP, by definition $\frac{dP}{P} = 0$

so $$\frac{dN}{N} = \frac{-dECOM}{ECOM}$$

Therefore, at Peak Power the %age increase in N RPM is just offset by the %age decrease in the combined efficiencies.

EC is shown in the Honda paper ref.(453) to be independent of N, provided that there is a suitable adjustment of ignition advance, so it is the decline of EV and EM which is significant in setting a Peak to Power.

Sample Power Curve

A sample Power Curve is shown on P.2, with the data made relative to the Peak Power (PP) point so as to illustrate the sensitivity of Power to Speed. In this example exceeding the RPM for PP by about 7% causes a drop of about 5% of Power.

COSWORTH HB3

NA; 75V8; B = 94mm x S = 63mm; B/S = 1.492; V = 3,498cc.

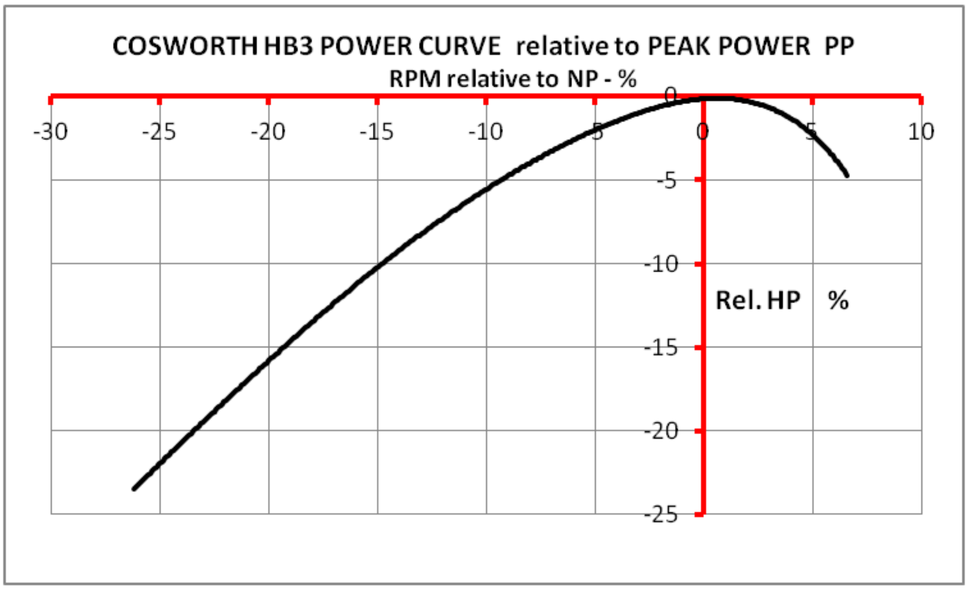
Start 1990

Peak Power (PP) =633 HP @ Peak Speed (NP) = 12,200 RPM.

BMPP = 13.3 Bar @ MPSP = 25.6 m/s

(See "3rd NA Era , Part 1")

								Datum		
RPM	9000	9500	10000	10500	11000	11500	12000	12200	12500	13000
Rel RPM%	-26.2	-22.1	-18	-13.9	-9.8	-5.7	-1.6	0	2.4	6.6
Power HP	484	517	547	576	599	617	630	633	630	603
Rel HP%	-23.5	-18.3	-13.6	-9	-5.4	-2.5	-0.5	0	-0.5	-4.7



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Deviation from SCF for 2000 Ferrari 049

The 2000 Ferrari 049 (Eg. 85) value for GS is 22% higher than the 47.4 average for the **I** group.
It is believed that this is associated with its high value of IVL/IVD at 0.384. It may be that some late 3rd NA Era engines are also faster than SCF suggests. The 1992 Honda RA122E/B (SO20, see Appendix 1) which had IVL/IVD = 0.315, was only 3.2% above 47.4 so a general change in IVL/IVD ratio may have occurred after that date.
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DST 27 January 2000
SCF.doc

Definition of SCF (Speed Correlation Factor)

Where:-

R = Compression Ratio:
B = Bore, mm;
S = Stroke, mm;
CN = No. of Cylinders;
VNI = No. of Inlet Valves per Cylinder;
IVD = Inlet Valve Head Diameter, mm;
IVL = Inlet Valve Maximum Lift, mm;
IOD = Inlet Valve Open Duration, (Valve Off/On Seat), Crank Degrees; at operating clearance, cold.
IVP = Inlet Charge Pressure at Valve, Atmospheres Absolute (ATA) (taken as 1 for Normally Aspirated);
VIA = Included Angle between Valves, Degrees;
IVA = Total Inlet Valve Head Area = CN x VNI x (Pi/4) x $(IVD/10)^2$ sq cm;
PA = Total Piston Area = CN x (Pi/4) x $(B/10)^2$ sq cm;
V = Total Swept Volume = PA x (S/10) cc;

and:-

NP = Crank RPM at Peak Power;

then **NP** is correlated in terms of **SCF**, which is simplified from a multiple regression anaysis of a sample of over 200 engines of all types and which has then been modified manually with two further factors by observation of variances, so that:-

$$SCF = \sqrt{\left[\frac{R \times (CN \times VNI \times Pi \times IVD) \times IOD \times (IVL \times IVP)^{1/3}}{V \times [\cos(VIA/2)]^{1/2}}\right]} \times YA \times YB$$

and

$$YA = 3.14 + (2/3) \times \left[\frac{R \times VIA}{1000}\right] - 1.1 \times \left[\frac{R \times VIA}{1000}\right]^2$$

YA has a maximum at (R x VIA) = 303

$$YB = 3/4 + 2.1 \times \left[\frac{IVA}{PA}\right] - 3.8 \times \left[\frac{IVA}{PA}\right]^2$$

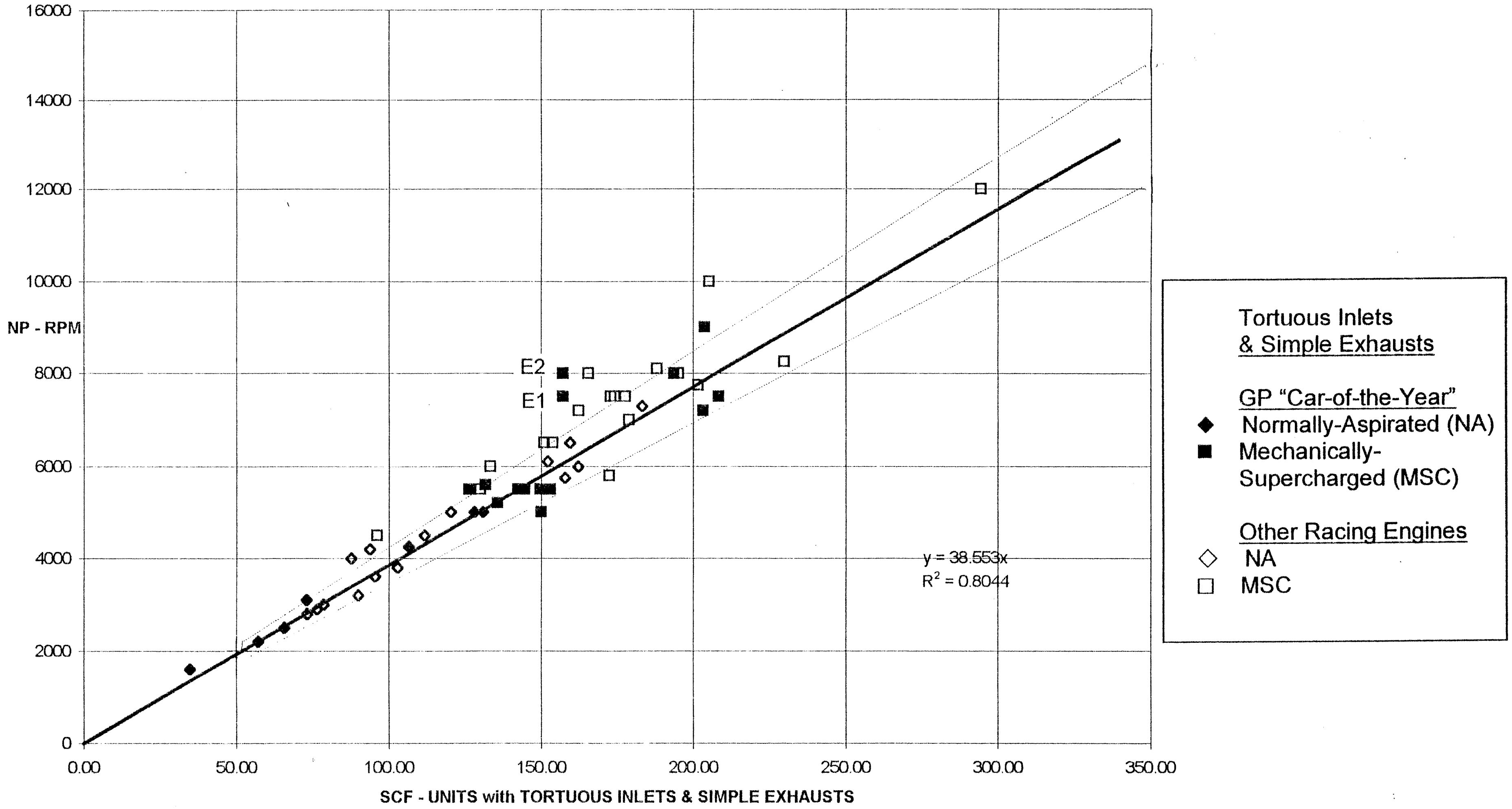
YB has a maximum at (IVA/PA) = 0.276, although results suggest the true maximum should be 0.32.
The equation for **YB** does not apply below (IVA/PA) = 0.1, since it should be 0 when (IVA/PA) = 0.

Note that the term (CN x VNI x Pi x IVD) = TIVC = Total Inlet Valve Head Circumference mm.

The Dimensions of **SCF** are:- $1/(\text{mm}^{5/6} \times \sqrt{1000})$

The value of the Gradient **GS = NP/SCF** depends mainly on the quality of the Inlet and Exhaust Systems. The highest figure (around 47.4) is produced from Individual, Fully-Tuned tracts at inlet and exit from the cylinders and the lowest (around 38.6) from multiple cylinders fed from one carburettor by untuned and tortuous tracts and with an untuned exhaust system.

GRAND PRIX ENGINE DEVELOPMENT 1906-1998 NP v. SCF FIGURE 93/DST



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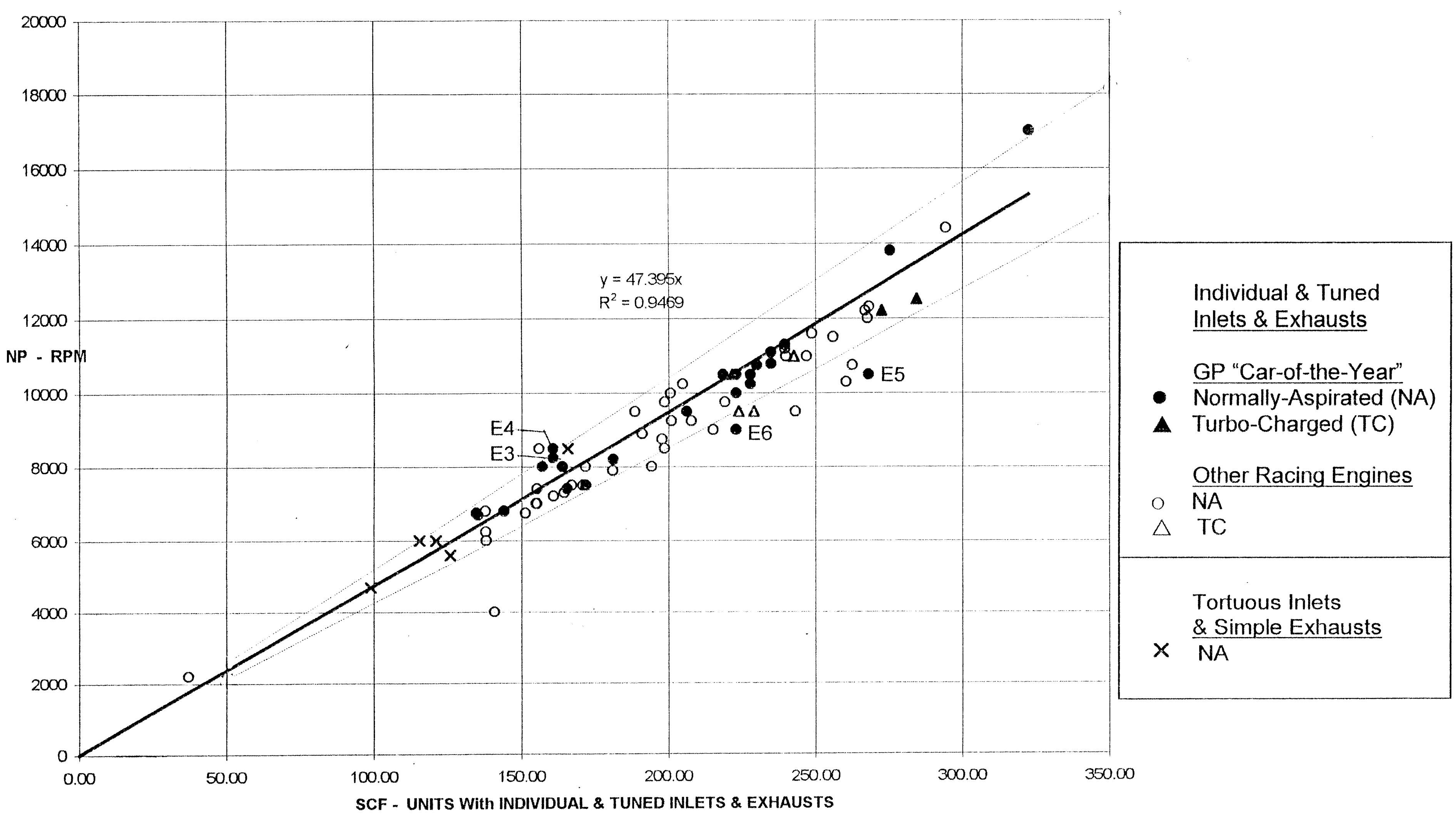

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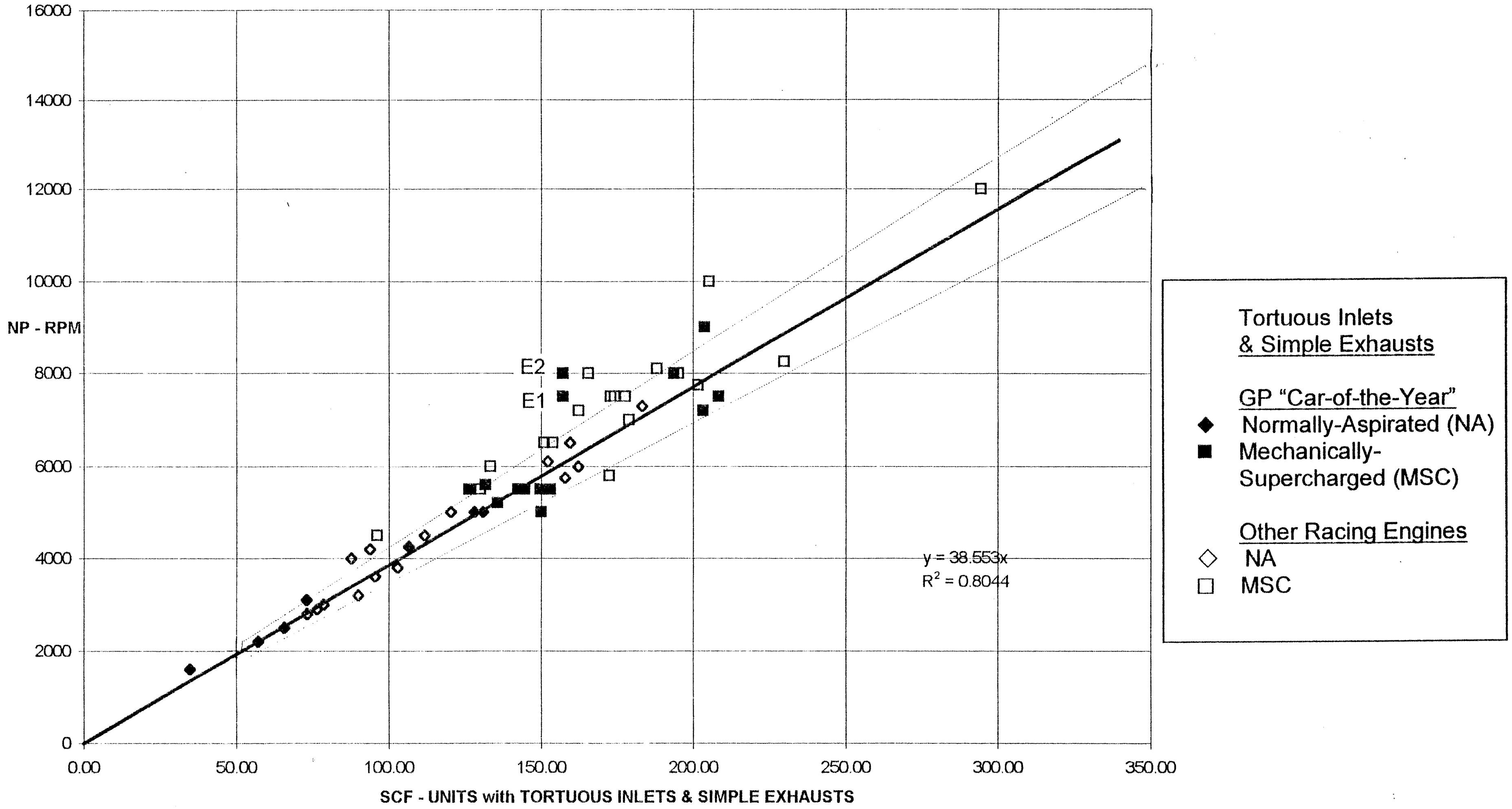

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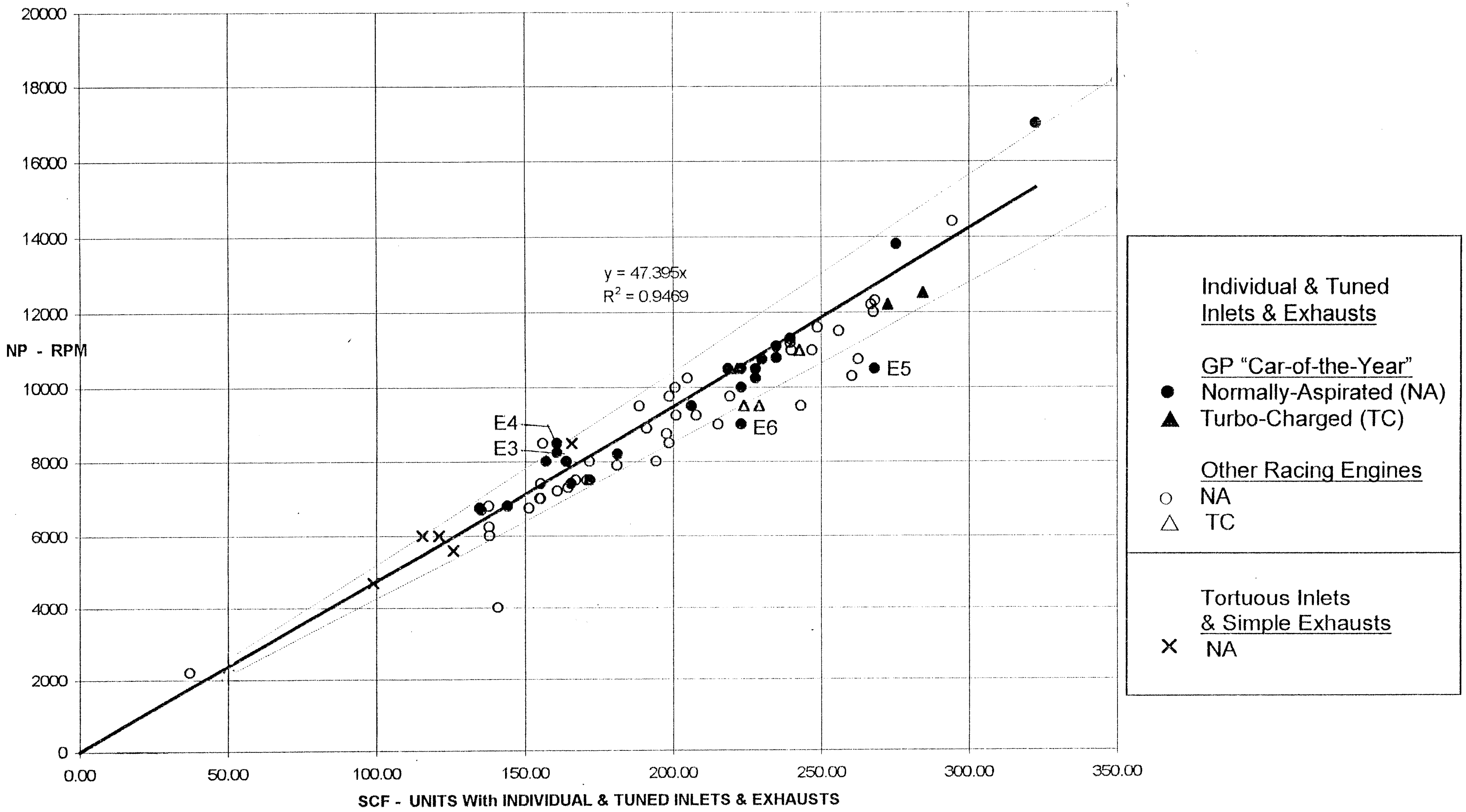


Note 13

MECHANICAL SPEED LIMITS

Part I : Piston Stressing

Part II : Piston-Ring Flutter as a limit to Piston Speed

Part III : Limiting Mean Valve Speed

This Note is an update and extension of theories discussed in two privately-issued papers by DST, as follows:-

Piston Engine Speed Limits	February 1987	(Ref. 748);
Optimum Bore/Stroke Ratio for Piston Engines	June 1991	(Ref. 749).

Optimum Bore/Stroke ratio

The consequence of mechanical speed limits to the “Bottom-End” of a piston engine (as discussed in Part I and Part II) and to the “Top-End” (as set out in Part III) is that there is an Optimum Bore/Stroke ratio to maximise volume-specific power at a chosen number of cylinders at a given state of Design and Material technology.

This Optimum is discussed in Note 21.

Note 13 Part I : Piston Stressing

From Newton's 2nd Law of Motion:-

$$\text{Acceleration} = \text{Force/Mass}$$

$$\therefore \quad \text{Acceleration} = (\text{Stress/Density}) \times (\text{Area/Volume}) \qquad \text{Eqn. 1.}$$

In the general case, Acceleration, Area and Volume are expressed in terms of a Characteristic Length and Time, so that:-

$$\{\text{Length}/(\text{Time})^2\} \propto (\text{Stress/Density}) \times \{(\text{Length})^2/(\text{Length})^3\}$$

resolving to:-

$$\textbf{Characteristic Speed} \propto \sqrt{(\textbf{Stress/Density})} \qquad \text{Eqn 2.}$$

Therefore, in the absence of other limits, the constructional materials available at a particular date will limit the speed of machinery, via the Stress permitted for a required fatigue life.

In a particular machine, the dimensions used to determine Acceleration, Area and Volume must be based on the critical parts. Practical experience is the guide here. In the highly-loaded – but also highly-successful – Cosworth type DFV V8 3L racing 4-stroke poppet-valve piston engine, designed originally in 1966 and developed steadily until 1982, it was found in the years of overhaul experience that parts lives were as follows (59):-

- Pistons (Al-alloy) — 500 racing miles (800 km) – About 2 races plus some practice
- Valve Springs (Steel Coils) — ." ." ."
- Valves (Steel) — 4 or 5 races
- Valve Spring Retainers — 7 or 8 races
- Bearings (Copper-Lead) — Full season (16 races)
- Connecting Rods (Steel) — Apparently no life limit
- Crank (Steel) — i.e. parts sized by stiffness required, not by stress

The *piston* and *valve-spring* lives therefore determined the routine rebuild time for the DFV engine. Engines prior to the DFV and post-WW1, particularly when supercharged, or post-DFV when turbocharged, also had the piston as the limiting component of the "Bottom-End". By the '90's, when normally-aspirated again, the competition was so fierce that pistons were being scrapped after only a few laps at Qualification power rating or one race with a fresh engine at "400km" race rating (419).

Consequently, where the "Bottom-End" mechanical limit on speed is concerned, it is the stressing of the piston which should be considered in the more detailed analysis (the mechanical limits which may be imposed alternatively or simultaneously by piston-ring flutter or the "Top-End" – the valve gear - are considered elsewhere).

The Theoretical Case

In the simplest analysis, considering the piston as the critical part:-

$$\text{Area} \propto (\text{Bore})^2 \propto B^2$$
$$\text{Volume} \propto (\text{Bore})^2 \times \text{Piston Height} \propto B^2 . PH$$

The Maximum Acceleration of the piston (MPD) is towards the crank at Top Dead Centre, as it is brought to rest and restarted in the opposite direction, and is given closely by:-

$$\text{MPD} \propto \text{Stroke} \times (\text{RPM})^2 \times \{ 1 + \text{Stroke}/(2 \times \text{Connecting-Rod Length-between-centres})\}$$

or

$$\text{MPD} \propto S . N^2 . \{ 1 + 1/(2 . CRL/S)\}$$

Entering these functions into Eqn 1. :-

$$S . N^2 . \{ 1 + 1/(2 . CRL/S)\} \propto (\text{Stress/Density}) \times B^2/(B^2 . PH)$$

$$(PH . S) . N^2 . \{ 1 + 1/(2 . CRL/S)\} \propto (\text{Stress/Density}) \text{ of the piston material}$$

The term { 1 + 1/(2 . CRL/S)} is fairly constant. An average number for CRL/S in racing engines from 1910 to 1998 is 2 and the variation of the term from CRL/S = 1.5 to 3 is +7% to –7% about the level at CRL/S = 2. This is the variation of stress at a given RPM, and the permitted variation of RPM at a given stress would be half the above percentages and of the opposite sign. Therefore, the expression may be treated as :-

$$(PH . S) . N^2 \propto (\text{Stress/Density}) \text{ of the piston material} \qquad \text{Eqn. 3.}$$

Assumptions regarding piston dimensional ratios.
(a): if the design of pistons leads to:-

$$PH \propto S$$

Then Eqn 3.. reduces to:-

$$S . N \propto \sqrt{(\text{Stress/Density})}$$

Or introducing Mean Piston Speed (MPS) = 2 . S . N

$$\mathbf{MPS \propto \sqrt{(Stress/Density)} \text{ of the piston material}} \qquad \text{Eqn. 4.}$$

(b): if the design of pistons leads to:-

$$PH \propto B$$

which could be the case if it is necessary to keep the skirt side-thrust rubbing pressure constant, or to keep the angularity of the piston rings constant, at some limiting values,

Then Eqn 3 becomes:-

$$\sqrt{(B . S)} . N \propto \sqrt{(\text{Stress/Density})}$$

also written as

$$\mathbf{\sqrt{(B/S)} . MPS \propto \sqrt{(Stress/Density)} \text{ of the piston material}} \qquad \text{Eqn. 5.}$$

to show more clearly how the simple "Mean Piston Speed Theory" of Eqn 4 is amended. This is known generally as the "Lanchester Theory", although it was not stated explicitly as such by that pioneer in 1906, but has been *derived* from his choice of an expression

$$\text{Power} \propto B^{1..5} . S^{0.5} \quad \text{which is given on p.164 of (369).}$$

Note that Lanchester made the condition that this would apply "given adequate port areas" (p.158 of 369) – a requirement often not met in subsequent designs (nowadays one would put "adequate valve-opening area x time").

The Practical Case

The Piston Ratios PH/B and PH/S have been examined over nearly 100 years of engine development on Figures 98/DST and 99/DST respectively. From 1906 to1955 it could be said, crudely, that PH/B was constant, averaging around 1.1, although with much scatter. From 1955, however, there has been a steady decline to about 0.5 in modern times. Similarly, the tendency in this latter period has been to reduce PH/S from about 1.1 to 0.9, having risen from about 0.7 in the preceding period. Again crudely, *if no other internal data are available*, it is probably better to assume for stress comparative purposes that PH/S is constant over the whole period at about 0.9, since PH/B certainly is *not* constant, i.e. Eqn 4 should be used rather than Eqn 5.

These two charts emphasise really that it is *not safe* to assume a fixed ratio of PH to either B or S for general conclusions about stressing limits – it should be included as a dimension in its own right. This is true particularly through the change since 1967 from high-compression 2-valve hemispherical heads with high piston crowns to the flat-topped pistons of lower mass permitted by 4-valve heads.

An analysis of Al-alloy racing piston weights (to use the usual term, although mass is the correct description) versus B and PH is given on Figure 97/DST. This includes slipper pistons (intended to reduce friction by cutting-down rubbing area) as well as full-skirted designs (Note I). The range of B is 50 to 110 mm. It will be seen that there is a strong correlation of:-

$$\text{Total Piston Weight (WPT)} \propto B^{1.5} .PH \qquad \text{Eqn. 6.}$$

where WPT is the whole purely-reciprocating assembly, including rings and gudgeon pin. The average error is 6% over 29 examples. Inability to scale-down over the range means that dimensional consistency is not followed in practice – a common situation in mechanical engineering, and one which Lanchester in 1906 had not sufficient examples to have been able to observe where internal combustion engines were concerned. *Density* is implicit on the RHS of Eqn 6.

Returning to Newton's 2nd Law and inserting Eqn. 6 gives:-

$$S \,.\, N^2 \,.\, \{ 1 + 1/(2 \,.\, CRL/S)\} \propto (\text{Stress x Area})/(\text{Density x } B^{1.5} \,.\, PH)$$

Adopting, as before, the simplification that { 1 + 1/(2 . CRL/S)} is nearly constant and that Area over which Stress applies $\propto B^2$:-

$$\mathbf{[(PH \,.\, S)/B^{0.5}] \,.\, N^2 \propto \text{Stress/Density of the piston material}} \qquad \text{Eqn. 7.}$$

or

$$\mathbf{[(PH/S)^{0.5}/B^{0.25}] \,.\, MPS \propto \sqrt{(\text{Stress/Density})} \text{ of the piston material}} \qquad \text{Eqn. 8.}$$

The LHS of Eqn. 7 has been titled "Piston Stressing Factor" (PSF). This has been calculated with PH, S and B in cm and N = NP, the Peak Power Speed, divided by 1000 to give a handy number (PSF has dimensions ($cm^{1.5}$ / $min^2 \cdot 10^6$)). PSF is plotted for racing engines against date on 100/DST.

Discussion of Piston Stressing Factor v. Date

PSF would be expected to rise over the years as materials of higher Stress/Density ratio became available (a general review of piston material development is given separately). Very broadly this is true, by a multiplication of 4 from 1914 to 2000. It seemed to take a few years, post WW1, before the change from cast-iron or steel pistons to Al-alloy, around 1914, had a significant effect on PSF, despite all the work done on the latter material in the War. There is a large amount of scatter, some of which is understood, i.e. some low points are push-rod (PROHV) or single-overhead camshaft (SOHC) units where the valve-gear or valve-area were limiting before the piston; some of the high points are believed to be "flash" readings (egs. Peugeot 1910 Voiturette, Delage 1926-1927 GP) or "Shelsley Sprint" or "Short Race" ratings (1936 Austin 750, 1954 BRM T15 Mk2: Note II). There is a group of Coventry-Climax engines (FPF, FWMV, FWMW) over 1959-1966 where Hassan deliberately used a valve timing which favoured mid-range torque at the expense of peak RPM (and power), and so limited peak piston stress (Note III).

The points given above being acknowledged, there is something of a dip in PSF after 1957. A possible explanation for this is that it coincided with the banning of alcohol fuel in favour of petrol. The high latent heat of evaporation of alcohol, compared with petrol, had been used since 1924 not only to cool the charge but also, as inlet pressures rose, with very-high %age alcohol content and excessive fuel/air ratio, to cool the pistons (607,468,31). To a lesser extent such methods were continued in 1954-1957 with normally-aspirated engines (Note IV). It would appear that, after the reversion to petrol, it took some time to get back to high levels of PSF, and turbocharging, even with oil-gallery-cooled pistons, delayed the recovery. The levels reached by 2000 indicate the increasing use of under-piston oil-spray cooling and reduction of life down to only 400km, i.e., pistons scrapped after only a single race.

Conclusion regarding "Rating Rules"

When Lanchester produced his 1906 paper he hoped to provide a simple rule from mechanical considerations which could be used to rate engines for power. The data given above emphasise that there is *no* simple way to predict speed from a mechanical "Bottom-End" limit across a substantial time period, because of variations in (at least):- valve-gear architecture; breathing & burning capability; piston (& ring) materials, proportions, heat-loading & cooling; and especially in the life required.

A "Rating Rule" may be found for a particular maker's design philosophy over a few years (e.g. Bastow's modified Lanchester formula for Climax engines (50)), but it is then found to be inaccurate for preceding and succeeding units.

Notes

I. The results show that slipper pistons do not in themselves reduce piston weight, because substantial inner supports are still required for the gudgeon-pin bearings, which would otherwise be carried by the full skirts.

II. The 1936 Austin 750 PSF = 1696 shown is at 10,000 RPM. For long races the speed was limited to 7,500 RPM, i.e. 44% lower stress. The 1954 BRM PSF = 1655 is at 12,000 RPM, which was very rarely reached.

III. The FPF 2.5L was a unit where CRL/S = 1.44 only , because of the way it had been enlarged from 1.5L. The stress therefore would have been 7.8% higher than at CRL/S = 2.

IV. It is interesting that the 1954-1955 Mercedes M196 (PSF = 1478 and 1569 respectively) only had one type of repeated fault – a car with a piston failure in the 1st race and a car with the same thing in the last event entered.

Note 13 Part I
Sub-Note A : Some Piston Examples

Photographs

An attached Figure gives a comparison of 3 forged Al-alloy pistons, all for NA high-compression (petrol-fuelled) engines, which illustrates the great reduction in piston mass accomplished between 1961 and 1996 by using under-piston oil-spray cooling and by also accepting life reduction from perhaps a whole season down to a single race.

The pistons shown are:-

	Date	Make	Type	v/c	PH	PH/B	PH/S
		CN x B x S mm	**NP RPM**		mm	**PSF**	
		WPT g	WPT/((B)$^{1.5}$.PH) Difference from correlation				
Top LHS:	1961	Norton	500cc Manx	2	81	0.94	0.95
		1a/c 86 85.62 =	**497cc 7,000**			**1,159**	
		592	2.90 +4.6%				
Middle RHS & Bottom RHS	1983	Honda	F2 RA263E	4	61	0.68	1.17
		80V6 90 52.3 =	**1996cc 12,000**			**1,531**	
		469	2.85 +2.7%				
Top, Middle & Bottom	1996	Yamaha-Judd	OX11A	4	42	0.47	0.89
		72V10 90 47.1 =	**2996cc 16,000**			**1,688**	
		273	2.41 -13.1%				

(The photos also show an OX11A Ti-alloy inlet valve, IVD = 37.5 mm)

These pictorial examples demonstrate how the controlling design parameter during this 35 year period was:-

$$\mathbf{PH} \quad \infty \quad \mathbf{S}$$

at a value around **1**, as shown by Fig. 99/DST.
(All 3 pistons are plotted on 97/DST but only the Yamaha on 98, 99, 100/DST).

Scale Drawings

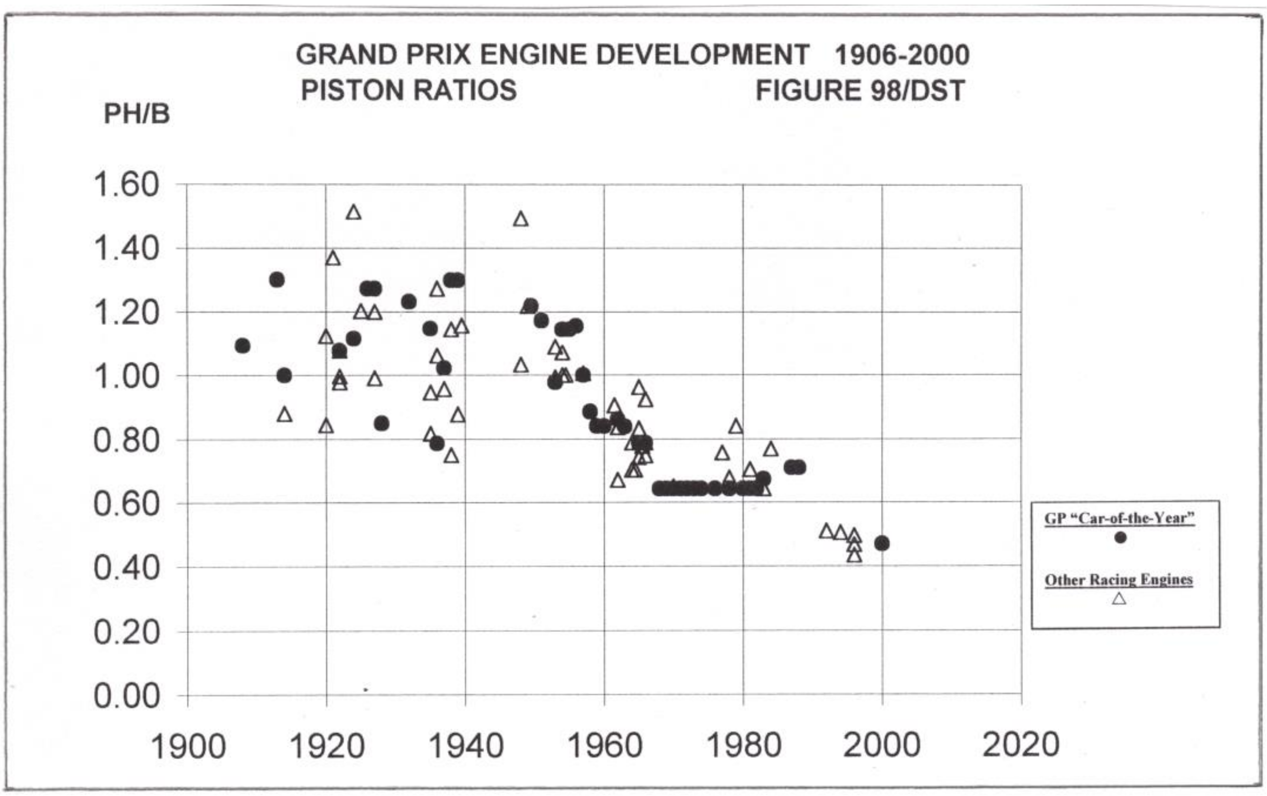
Two scale drawings are attached which give full details of:-

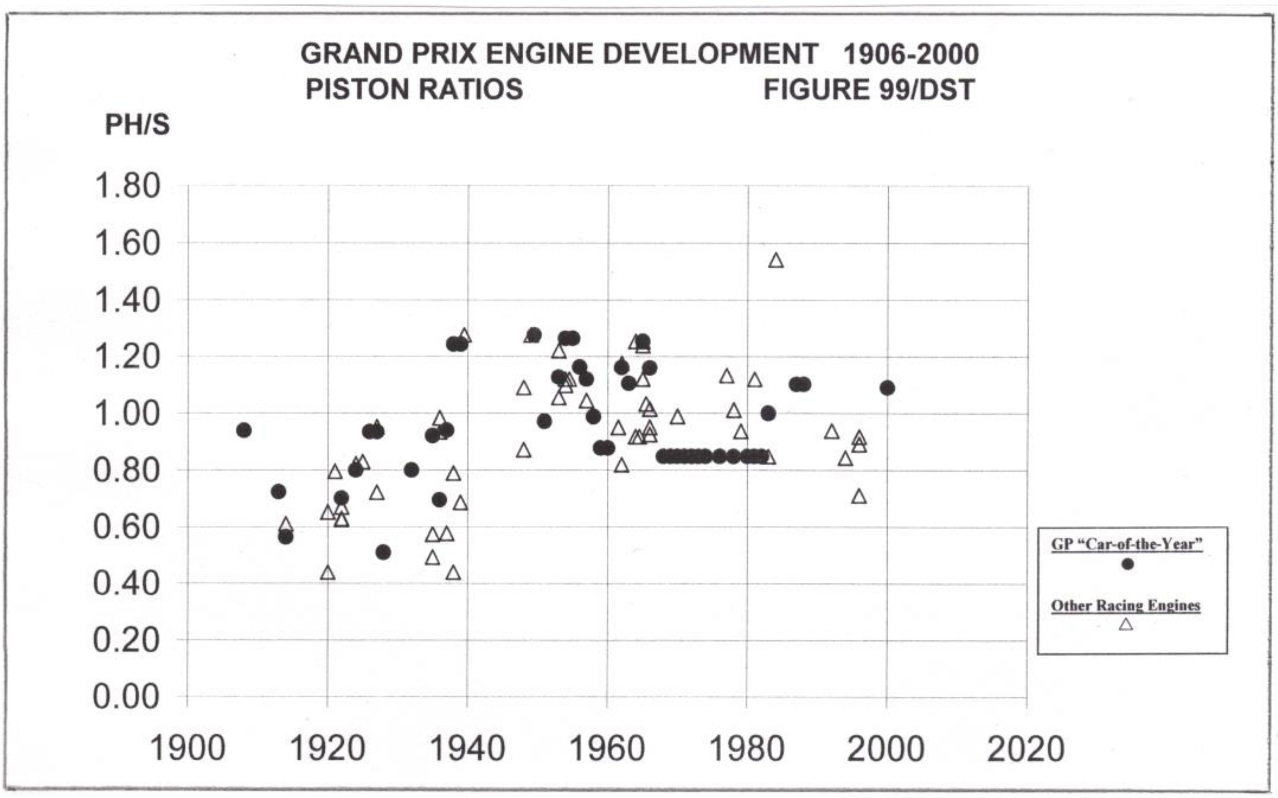
	PSF	WPT/((B)$^{1.5}$.PH)	Diff. from Corrln.
• Yamaha-Judd OX11A (DASO 674)	1,688	2.41	-13.1%
• Mugen-Honda MF301 (DASO 672)	1,318	2.82	+1.8%

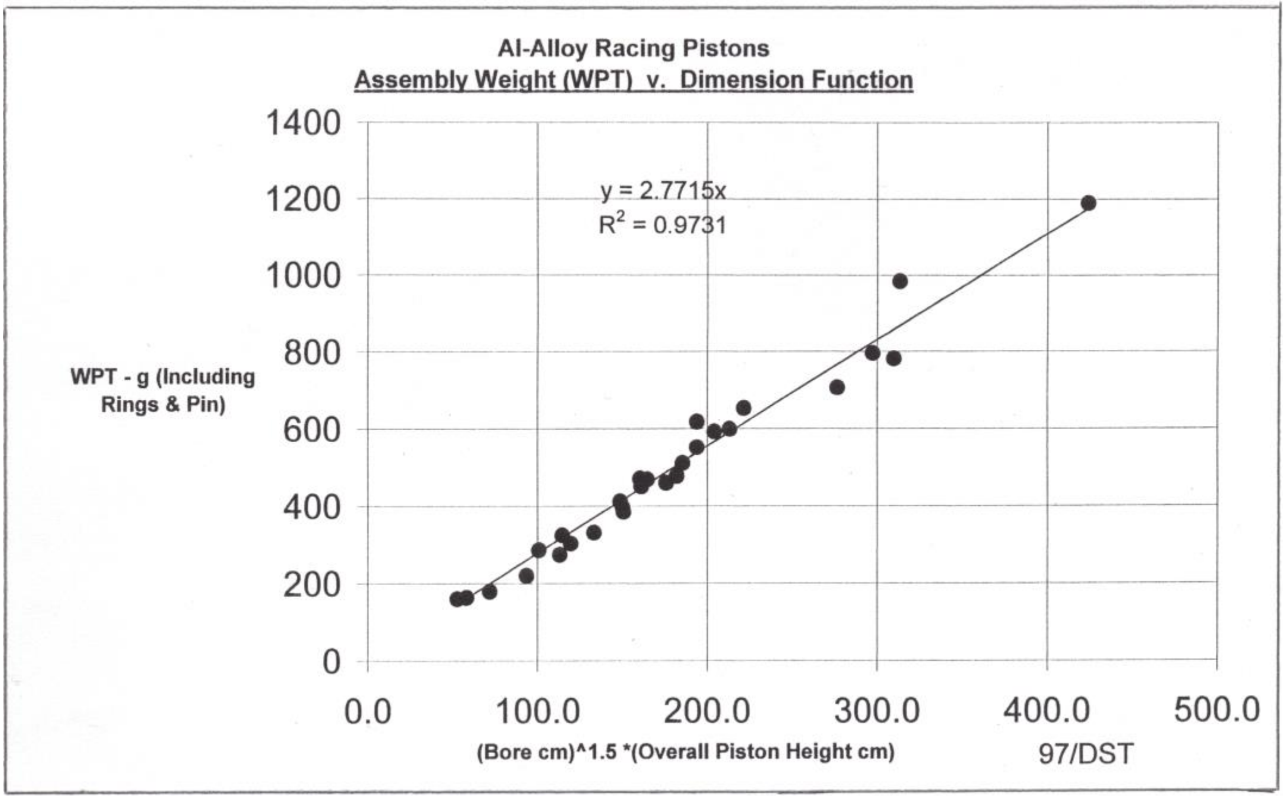
Both are 1996 pistons (and both are on 97/DST).

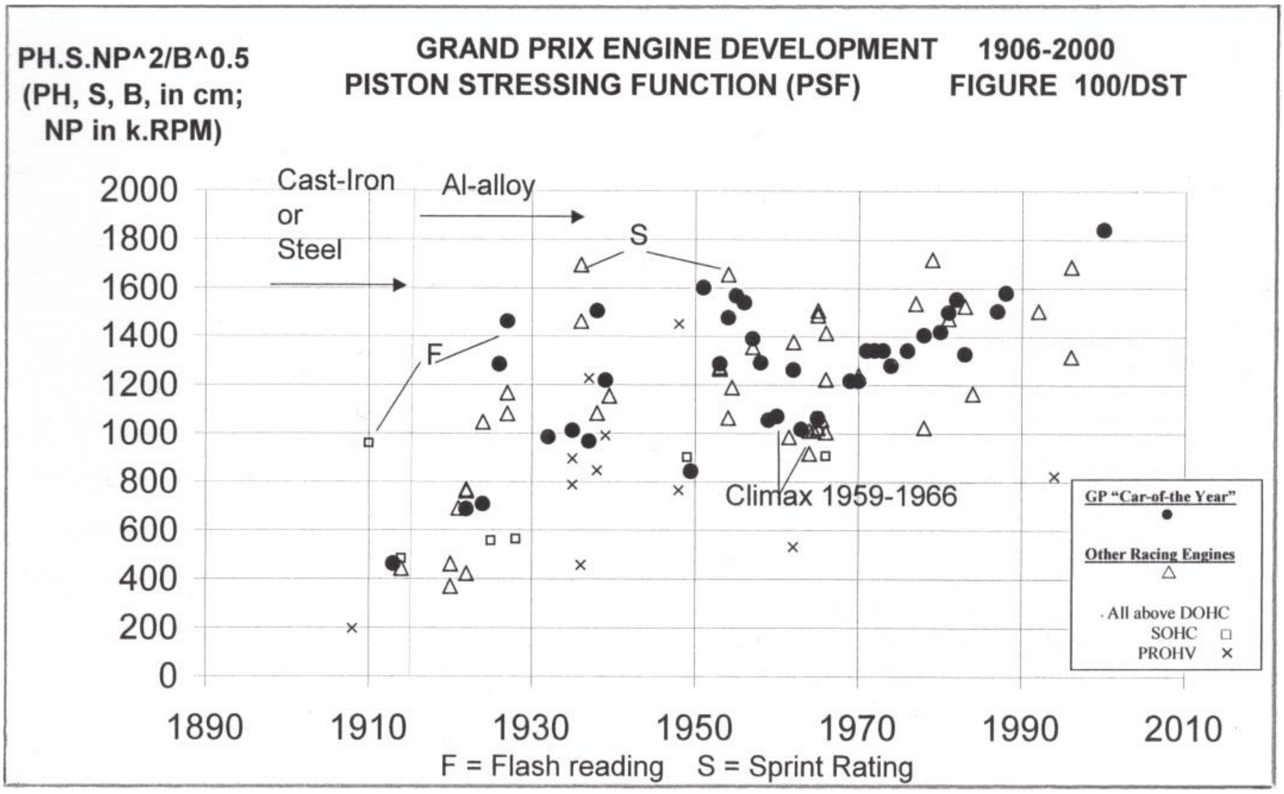
The Mugen-Honda is particularly interesting because the drawing was made of a piston taken from the winning engine at Monaco in 1996 (mounted with its Ti-alloy con-rod and given by Socheiro Honda to Mr Alfred Briggs of Derby as a mark of appreciation for many years service with Honda).

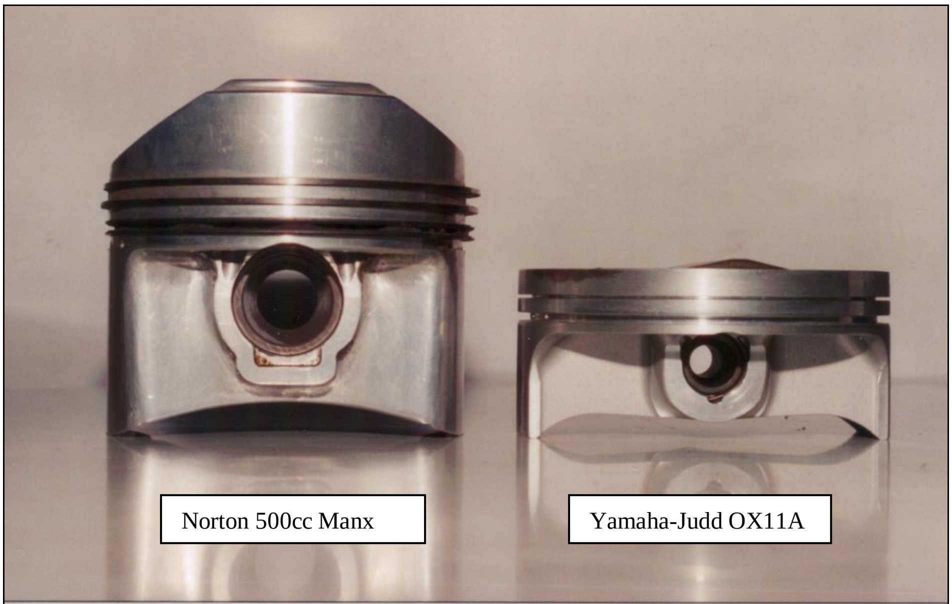
Note that the Mugen B = 93 mm and CRL/S = 2.72 were both unusually high for the period, because the engine was adapted from 3.5L to 3L in 1995 by a stroke reduction, so as to retain the existing crankcase. Compared to CRL/S = 2 the actual con.-rod dimensions would have reduced piston stress by about 5%.

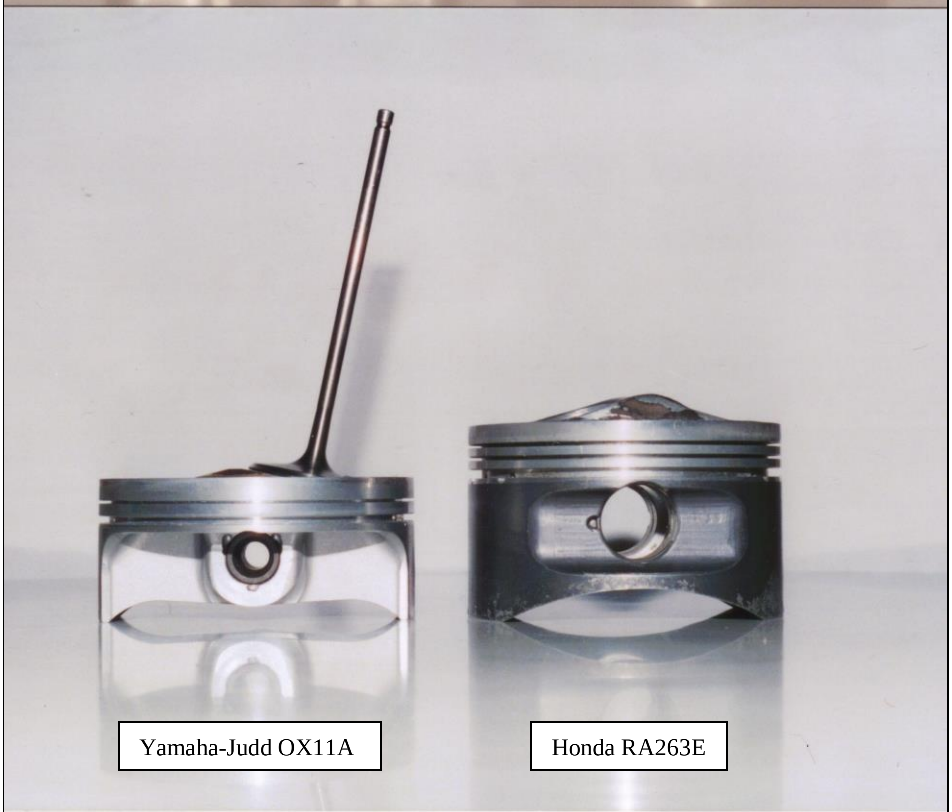

GRAND PRIX ENGINE DEVELOPMENT 1906-2000
PISTON RATIOS
FIGURE 98/DST
PH/B
1.60
1.40
1.20
1.00
0.80
0.60
0.40
0.20
0.00
1900
1920
1940
1960
1980
2000
2020
GP "Car-of-the-Year"
Other Racing Engines

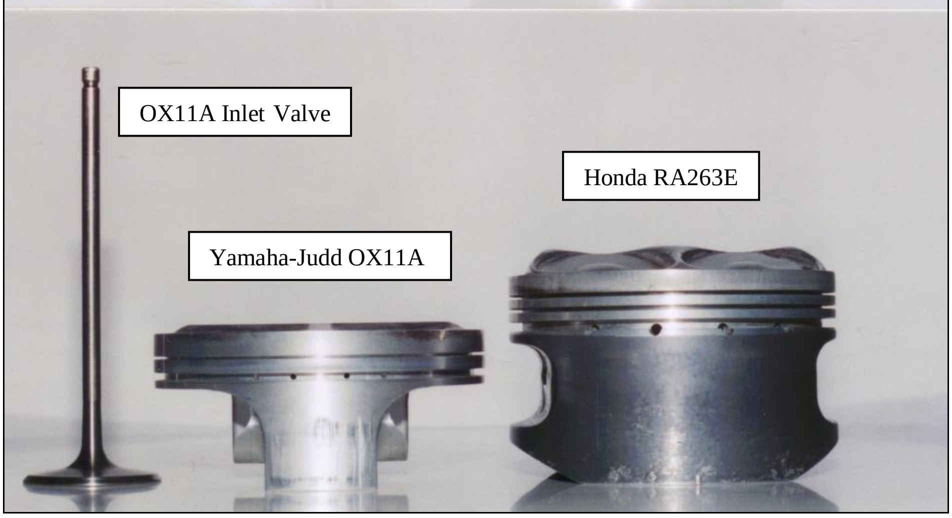

GRAND PRIX ENGINE DEVELOPMENT 1906-2000
PISTON RATIOS
FIGURE 99/DST
PH/S
1.80
1.60
1.40
1.20
1.00
0.80
0.60
0.40
0.20
0.00
1900
1920
1940
1960
1980
2000
2020
GP "Car-of-the-Year"
Other Racing Engines

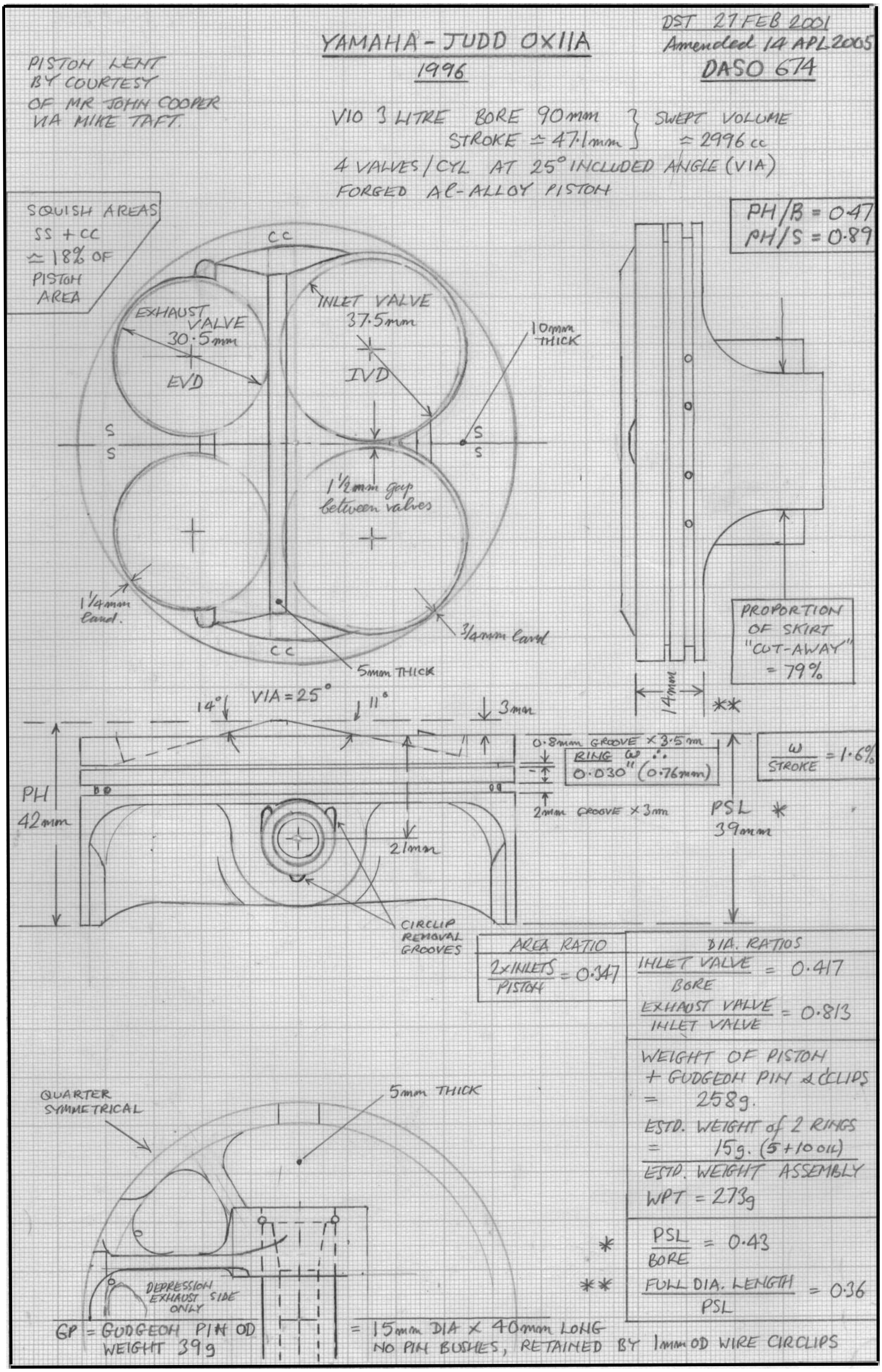

Al-Alloy Racing Pistons
Assembly Weight (WPT) v. Dimension Function
y = 2.7715x
R^2 = 0.9731
WPT - g (Including Rings & Pin)
1400
1200
1000
800
600
400
200
0
0.0
100.0
200.0
300.0
400.0
500.0
(Bore cm)^1.5 *(Overall Piston Height cm)
97/DST

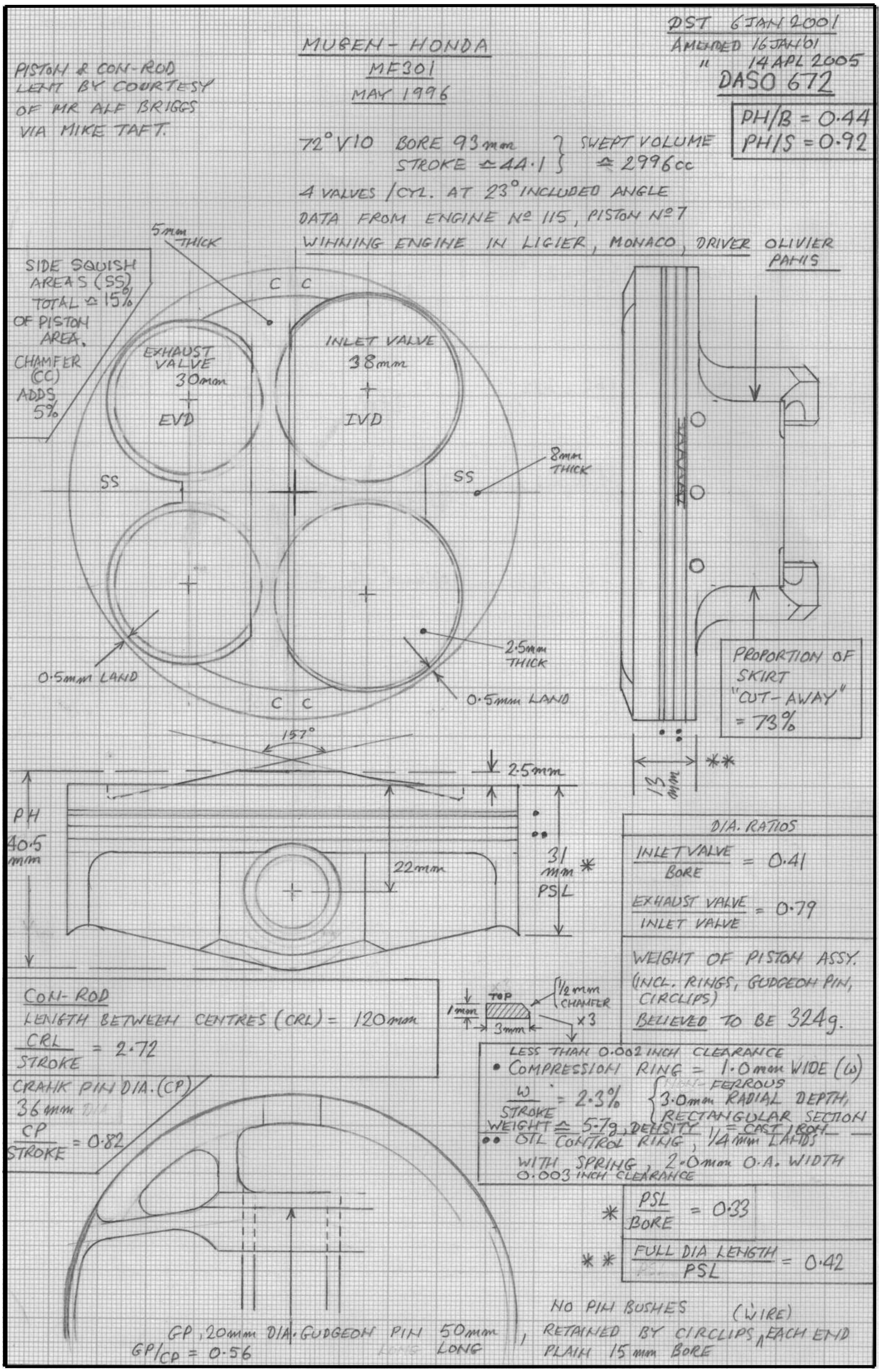

PH.S.NP^2/B^0.5
(PH, S, B, in cm; NP in k.RPM)
GRAND PRIX ENGINE DEVELOPMENT 1906-2000
PISTON STRESSING FUNCTION (PSF) FIGURE 100/DST
Cast-Iron or Steel
Al-alloy
S
F
Climax 1959-1966
2000
1800
1600
1400
1200
1000
800
600
400
200
0
1890
1910
1930
1950
1970
1990
2010
GP "Car-of-the Year"
Other Racing Engines
All above DOHC
SOHC
PROHV
F = Flash reading S = Sprint Rating


Norton 500cc Manx
Yamaha-Judd OX11A


Yamaha-Judd OX11A
Honda RA263E


OX11A Inlet Valve
Honda RA263E
Yamaha-Judd OX11A


YAMAHA - JUDD OX11A
1996
DST 27 FEB 2001
Amended 14 APL 2005
DASO 674
PISTON LENT
BY COURTESY
OF MR JOHN COOPER
VIA MIKE TAFT.
V10 3 LITRE BORE 90mm
STROKE ≃ 47·1mm
SWEPT VOLUME
≃ 2996 cc
4 VALVES/CYL AT 25° INCLUDED ANGLE (VIA)
FORGED Al-ALLOY PISTON
SQUISH AREAS
SS + CC
≃ 18% OF
PISTON
AREA
PH/B = 0·47
PH/S = 0·89
CC
EXHAUST
VALVE
30·5mm
EVD
INLET VALVE
37·5mm
IVD
10mm
THICK
S
S
S
S
1½mm gap
between valves
1¼mm
land.
¾mm land
CC
5mm THICK
PROPORTION
OF SKIRT
"CUT-AWAY"
= 79%
14mm
**
VIA = 25°
14°
11°
3mm
0·8mm GROOVE × 3·5mm
RING ω ∴
0·030" (0·76mm)
2mm GROOVE × 3mm
ω / STROKE = 1·6%
PH
42mm
21mm
PSL *
39mm
CIRCLIP
REMOVAL
GROOVES
AREA RATIO
2×INLETS / PISTON = 0·347
DIA. RATIOS
INLET VALVE / BORE = 0·417
EXHAUST VALVE / INLET VALVE = 0·813
WEIGHT OF PISTON
+ GUDGEON PIN & CLIPS
= 258g.
ESTD. WEIGHT of 2 RINGS
= 15g. (5+10 oil)
ESTD. WEIGHT ASSEMBLY
WPT = 273g
QUARTER
SYMMETRICAL
5mm THICK
DEPRESSION
EXHAUST SIDE
ONLY
* PSL / BORE = 0·43
** FULL DIA. LENGTH / PSL = 0·36
GP = GUDGEON PIN OD = 15mm DIA × 40mm LONG
WEIGHT 39g
NO PIN BUSHES, RETAINED BY 1mm OD WIRE CIRCLIPS



MUGEN - HONDA
MF301
MAY 1996
DST 6 JAN 2001
AMENDED 16 JAN 01
" 14 APL 2005
DASO 672
PH/B = 0.44
PH/S = 0.92
PISTON & CON-ROD LENT BY COURTESY OF MR ALF BRIGGS VIA MIKE TAFT.
72° V10 BORE 93mm
STROKE ≏ 44.1
SWEPT VOLUME ≏ 2996cc
4 VALVES / CYL. AT 23° INCLUDED ANGLE
DATA FROM ENGINE Nº 115, PISTON Nº 7
WINNING ENGINE IN LIGIER, MONACO, DRIVER OLIVIER PANIS
SIDE SQUISH AREAS (SS) TOTAL ≏ 15% OF PISTON AREA.
CHAMFER (CC) ADDS 5%
5mm THICK
C C
EXHAUST VALVE 30mm
EVD
INLET VALVE 38mm
IVD
SS
SS
8mm THICK
2.5mm THICK
0.5mm LAND
0.5mm LAND
C C
PROPORTION OF SKIRT "CUT-AWAY" = 73%
13 mm
**
157°
2.5mm
PH 40.5 mm
22mm
31 mm PSL *
DIA. RATIOS
INLET VALVE / BORE = 0.41
EXHAUST VALVE / INLET VALVE = 0.79
WEIGHT OF PISTON ASSY. (INCL. RINGS, GUDGEON PIN, CIRCLIPS) BELIEVED TO BE 324g.
CON-ROD
LENGTH BETWEEN CENTRES (CRL) = 120mm
CRL / STROKE = 2.72
CRANK PIN DIA. (CP) 36mm
CP / STROKE = 0.82
TOP
1mm
3mm
1/2mm CHAMFER
x3
LESS THAN 0.002 INCH CLEARANCE
• COMPRESSION RING = 1.0mm WIDE (ω)
ω / STROKE = 2.3%
NON-FERROUS
3.0mm RADIAL DEPTH, RECTANGULAR SECTION
WEIGHT ≏ 5.7g, DENSITY = CAST IRON
•• OIL CONTROL RING, 1/4mm LANDS WITH SPRING, 2.0mm O.A. WIDTH
0.003 INCH CLEARANCE
* PSL / BORE = 0.33
** FULL DIA LENGTH / PSL = 0.42
NO PIN BUSHES
GP, 20mm DIA. GUDGEON PIN 50mm LONG, RETAINED BY CIRCLIPS (WIRE) EACH END
PLAIN 15mm BORE
GP/CP = 0.56

Piston Ring Flutter as a limit to Piston Speed **DST 11 February 2003**

Up to the early 50s piston acceleration – or rather *de*celeration of the piston-rings at the end of the exhaust stroke – could form a limit to engine speed. This was because the ring momentum at the minimum producible rectangular-section (plain) ring axial width (and therefore minimum mass) could overcome the residual gas pressure so that the ring was thrown to the top of its groove. Gas pressure then could no longer reach the inside of the ring to compensate for that on the outside and the ring would be forced inwards. Apart from the resulting gas blow-by, losing power, degrading the oil and pushing it out of the engine, the ring radial vibration with RPM – "flutter" – would lead to very rapid fatigue failure of the cast-iron material then used. Over-revved engines would be found to have their rings broken into many pieces (see Sub-Note A and sketch on Page 5). Obviously, all the bad effects given above would then be magnified.

The basic relation

When a piston-ring is on the point of "topping-out" in its groove, then, by Newton's 2nd Law:-

(Ring Mass) x (Maximum Piston Deceleration, MPD)
= (Pressure Difference across Ring, Δp) x (Ring plan area)
and (Ring Mass) = Ring(Density,DR) x (Axial Width,w) x (plan area)
so

$$DR \,.\, w \,.\, MPD = \Delta p$$

w.MPD = Δp/DR at flutter onset

The Hepworth relation

The piston and ring manufacturer J.Hepworth must have discovered that Δp does not vary greatly between engines at the critical condition because he published detailed recommendations in June 1953 which could be translated into:-

w.MPD = Constant = 5000 inch . ft/sec^2 (say, 4000 mm.g)

for plain cast-iron rings at flutter onset (745).

The smallest width which could be made at that date was 3/64'' (1.2mm), which set a limiting value of MPD at 106,000 ft/sec^2 (32,210 m/sec^2 or 3300g). This was misunderstood sometimes as an *absolute* limit, instead of a figure which would rise when technology permitted thinner rings.

Ring geometry and piston speed

By substituting:-

$$MPD = [(Stroke,Smm) \times (RPM,N)^2 \times (1 + 1/(2 \,.\, CRL/S))]/(1.789 \,.\, 10^6) \quad g$$

where CRL = Connecting-Rod Length between bearing centres

and Mean Piston Speed, MPS = (Smm . N)/30,000 m/s

and using an average value of CRL/S = 2, then the Hepworth relation (**w.MPD = 4000 mm.g**) can be converted into:-

MPS.√(w/S) = 2.5 m/s nearly enough.

Therefore, the limits to MPS set by a satisfactory plain cast-iron piston-ring life are:-

w/S %	**4**	**3**	**2**	**1.5**	**1**
MPS m/s	**12.5**	**14.4**	**17.7**	**20.4**	**25.0**

Experimental results

In Sub-Note B are given results for 9 engines, of a wide variety, in which the plain iron rings were known to be at or over the flutter onset.

The averages are:-

w.MPD = 4060 mm.g and **MPS.√(w/S) = 2.52 m/s**

Hepworth would have known of the data on 3 of these units when formulating his relation.

If the density of iron is entered into his relation, the value of **Δp** can be calculated as **44 psi (3 Bar**), regardless of whether the engine is Normally-Aspirated (NA) or Pressure-Charged (PC). This is above the exhaust pressure at Top Dead Centre where MPD is a maximum, but it does not approach the maximum cylinder pressure. It is probably the averaged end result of gas having to pass up and down the very-narrow annular clearance communicating from the combustion-chamber to the ring groove. The high value of **w.MPD** for the 1939 Mercedes-Benz in Sub-Note B may be a consequence of the top ring being rather low down on the piston (4) so that Δp is "trapped" at a higher-than-typical level.

For very short durations engines could be operated beyond the flutter boundary, e.g. the 1936 Austin 750cc listed in Note Ersatz 16 Part I was run up to 10,000 for Shelsley hill-climbs, a matter of only a few seconds at max. speed before the engine would be overhauled.

The Dykes' Ring

It is ironic that, by the date Hepworth published his recommendations, the invention by Prof. Dykes of the **L**-section ring in the late 40s (174) had already enabled designers to avoid the operational limits set by plain rings. The inner leg of the **L** was dimensioned to "top-out" first in a closer-clearance groove so as to permit the outer ring proper (the vertical part of the **L**) to retain its pressure backing and continue to seal the cylinder without self-destructive vibration (see sketch on Page 5). Naturally this was a very popular solution in the 50s and 60s in UK racing engines (see Sub Note C). Possibly the last car engine to use the type was the Ford-Cosworth FVA of 1966 (583) but the Dykes' ring was still fitted to some GP racing 2-stroke motor-cycle engines into the mid-80s (755)*.

In the literature of the 60s the Dykes' ring was often described as "pressure-backed", which was a misunderstanding of the basic ring operation since *all* rings are "pressure-backed" until topping-out occurs. However, because its pressure-backing was always assured, the Dykes' ring could be made with lower radial spring pressure and so reduced friction. Ref. (754) describes how this was carried to excess in the 1965 Climax and blow-by increased again so that Jimmy Clark nearly lost the British GP of that year by using up all the oil in his Lotus.

Thinner plain rings

As a less expensive alternative to the Dykes' ring, thinner plain rings were made from the early 60s onwards.An extreme example of this was used in the late 1964 Honda RC114 IL2 50cc engine. This had MPD around 7300g and, judging from a full-size photo in (75), w = 0.5mm for a product of about 3600mm.g. The Ford-Cosworth DFV 3L, introduced in 1967, never used Dykes' rings (746) and the 1983 development was able to run at 5650g at peak power with a plain top ring in stainless-steel (Mo-filled) reduced to 0.030'' (0.76mm)(746). The product **w.MPD** therefore was 4300mm.g for a material slightly less dense than iron. The value of **w/S** was 1.2% and **MPS.√(w/S)** = 2.62.

Engines of the 90s

Values of MPD continued to rise in GP engines through the 90s as B/S ratios were increased, but 'w' did not decline in inverse proportion. Two examples are known:-

Engine / Data Source	Smm	w mm	w/S	N / **w.MPD**	CRL/S	MPD / **MPS.√(w/S)**	MPS
1996 Mugen-Honda MF301 72V10 3L							
(672)	44.1	1	2.3%	15,000RPM	2.72	6565g	22.05m/s
				6565 mm.g		**3.32 m/s**	
1996 Yamaha-Judd OX11A 72V10 3L							
(674)	47.1	0.762	1.6%	16,000RPM	≅ 2.2	8271g	25.12m/s
				6303.mm.g		**3.20 m/s**	

While there is good agreement of the critical factors between these engines, both with plain rings, the Mugen ring known to be ferrous, they were running at **w.MPD** *60% higher* than the relationship derived from earlier units. The 1998 Ilmor-Mercedes-Benz FO110G is known to have run at 8500g (559) but 'w' is unknown. All these rings are mounted high up on the piston. Modern ring material is about 10% less dense than cast-iron (606), which adds that proportion to the limiting product **w.MPD**.

It is possible that, with improved materials, the rings are *allowed* to flutter and can still survive for a life required of only 400km and that the really-powerful sump-scavenging and de-aeration systems now used (69), with improved oil, still allow the lubricant to do a satisfactory job for the less-than-2 hours needed, despite blow-by.

Cases are known in recent years where piston-ring-flutter has been acknowledged: in the 1989 Canadian GP, run in rain, the Honda V10 3.5L engine of a McLaren failed when leading because on-off use of the throttle to keep control on the slippery track led to ring flutter and blow-by which caused excessive oil loss (727); the 1994 Peugeot V10 3.5L engine, also in a McLaren, lost all of its 15L of oil at Monaco (727), almost certainly because of ring-flutter since, not long afterwards, a McLaren-Peugeot caught fire on the British GP grid and it was admitted that ring-flutter had forced oil loss onto the exhaust (574).

There is a further possibility to explain the modern high **w.MPD** factors – that the engines now rotate in the normal running range so much faster than the natural frequency of radial vibration of the ring that it cannot resonate, i.e. flutter.In this situation, dropping the RPM, as might occur while waiting on the grid, at slow corners or in slippery conditions, could bring the ring into its flutter region. That would explain the Honda and Peugeot cases cited above.

---

*

2-stroke piston-rings cannot flutter, because the outward stroke is always under compression. The use of a Dykes' ring in such engines will have been to reduce friction compared with a plain ring having high radial pressure.

The Fiat – Cappa piston-ring solution

Looking back to the standard Fiat piston-ring arrangements of 1922-1927 provides a clue to another method of avoiding ring flutter. In these engines, at the initiative originally of Guilio Cappa, the rings were dimensioned so that they bottomed in their grooves before the piston touched the wall. There were *double* plain rings in each groove, each with w = 2.5 mm, 2 double sets per piston, plus double oil scrapers. Undoubtedly the intention was to reduce friction, the cylinders being steel and the

rings almost certainly cast-iron. Since the effective mass of the top ring was equivalent to 5 mm width, the operating **w.MPD** was far into the flutter region described above, but clearly no flutter occurred since both the 1922 and 1923 engines raced successfully at high speed over 800km at Monza. The values of **w.MPD** were 8500 and 9400 mm.g respectively (66).The 1927 engine, which won a short but fast race at Monza in that year ran at no-less than **14,000 mm.g** (66).

The explanation of these very high-but-innocuous values of **w.MPD** must lie in the inability of the rings to flutter radially at destructive amplitude because of their near-filling of the groove.

There is another piece of evidence concerning this type of ring-groove relationship. When the 1931 Rolls-Royce 'R' engine was in excessive oil consumption trouble, undoubtedly due to ring flutter (see Sub Note B), the Development Engineer, Cyril Lovesey, recorded afterwards:-

"We had been told by various piston ring manufacturers that considerable reduction in oil consumption could be obtained by rings having only a few 1000ths of an inch between them and the back of the ring groove – in other words the rings practically filling the groove – but we could find no appreciable benefit from this" (615).

This shows that, although at that date ring-flutter was not understood – that came about gradually during the mid 30s (e.g. 626) – a way of avoiding it *had* been deduced for some cases and quite likely from the Fiat experience, although not found effective in the "R" engine. It is not entirely clear how that unit's problems *were* solved (there being no reference to narrower rings), although extra scrapers and a larger sump were fitted and the consumption reduced to 1/10th of the early tests (615).

However, the groove-filling-ring solution appears to have been neglected post the Fiats as far as many racing engines were concerned, hence the limiting situations described in Sub Note B. The Mercedes-Benz M196I (300SLR) sports-racing engine is shown very clearly by a good section drawing in (468) to have *2 upper rings "chock" in their grooves*, which is *not* inaccurate draughtsmanship because the 2 lower rings are shown *with* radial clearance. The data for this unit are:-

Engine	Data Source	Smm	w mm	w/S	N	CRL/S	MPD	MPS
					w.MPD		**MPS.√(w/S)**	
1955 Mercedes-Benz M196I IL8 3L								
	(468)	78	2	2.56%	7,500RPM	1.86	3112g	19.50m/s
					6224 mm.g		**3.12 m/s**	

The 1954 M196 GP engine, from which the 300SLR was derived, had similar factors. It is *possible* – the data are not complete – that the pre-WW2 Mercedes engines were designed the same way.

It is known that, in the early 80s, highly-tuned Ford V8 7L stock-car engines, revving to 6,000 RPM although standard production 5/64'' rings had to be retained by the rules in an engine with S = 3.78'' so that w/S = 2.1% and MPS = 19.2 m/s so **MPS.√(w/S) = 2.76**, had *shims fitted behind the rings* to reduce radial clearance to only 0.005'' (1/10th % of the bore) (220). Clearly these were flutter-stoppers.

It may be that the tight ring-groove design was revived in the 90s in some GP engines as another way to beat the **w.MPD** limit. However, there is no hint of this in the literature as reviewed by the author and the Mugen-Honda MF301 quoted earlier had a ring of 3 mm radial depth in a 3.5 mm groove, i.e., *not* "chock".

Sub Note A

The famous "Motor Sport" correspondent Denis Jenkinson reported on the 1953 German GP practice as follows:-

"*…the Belgian Ferrari engine"* (IL4 Type500 2L) "*was spread all over the floor of its lock-up. It had lost a lot of power and when the block was lifted it was found that all the piston rings had disintegrated into tiny pieces, a source of wonder to the owners, but a sign of over-revving to the Ferrari mechanics*" (750).

Ref. (752) gives Harry Mundy's report on a tuned-up Ford Zodiac road-car engine (IL6 2.6L) in which higher RPM had been used regularly until excessive blow-by was noticed and :-

"*…on stripping the engine it was found that rings were broken in each of the pistons*".

See also Sub Note B for the factors of these engines.

Sub Note B

Experimental Results

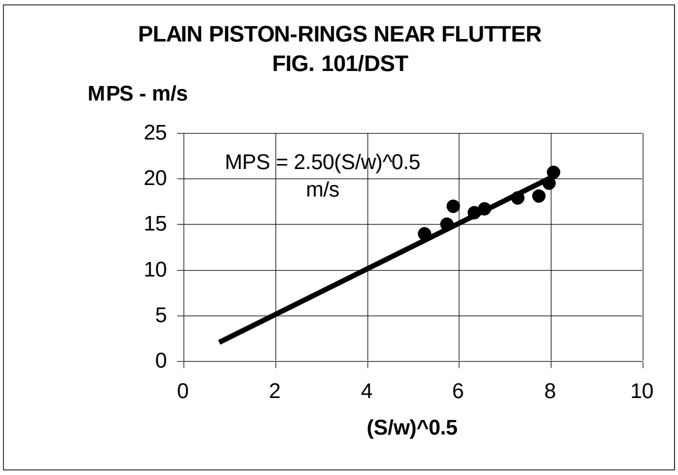
Data collected from engines in which the plain iron rings were known to be at or over the flutter limit have been tabled as follows (NA = Normally-Aspirated; PC = Pressure-Charged):-

Engine Data Source	Smm	w mm	w/S	N RPM	CRL/S	MPD g	MPS m/s
				w.MPD mm.g	**MPS.√(w/S) m/s**		
1. 1931 Rolls-Royce 'R' 60V12 36.6L PC - Early Tests							
615,766	167.64	2.778	1.66%	3200	1.706	1241	17.88
				3448	**2.30**		
2. 1936 Austin IL4 750cc PC - Long-Distance Rating							
197	65.09	1.5	2.30%	7600	1.951	2639	16.49
				3959	**2.50**		
3. 1939 Mercedes-Benz M163 60V12 3L PC							
4,30,468	70	2	2.86%	7200	2.257	2479	16.80
				4957	**2.84**		
4. 1950 BRM T15 135V16 1.5L PC – Early Tests – Before fitting Dykes' rings							
174	48.26	1.191	2.47%	10,000	2.172	3318	16.09
				3952	**2.53**		
5. 1951 Jaguar XK120C IL6 3.4L NA Early Tests – Before fitting Dykes' rings							
107,747	106	1.984	1.87%	5000	1.858	1880	17.67
				3729	**2.42**		
6. 1953 Ferrari Type 500 IL4 2L NA – at RPM permitted (This is the unit described in Sub Note A (750)) * It is assumed that w = the minimum then producible in the UK, i.e. 3/64inch							
80,749	78	1.191*	1.53%	7900	1.821	3468	20.54
				4132	**2.54**		
7. 1954 Coventry Climx FWA IL4 1100cc NA – Early Tests – Before fitting Dykes' rings							
132,133	66.675	2.381	3.57%	6200	1.8095	1828	13.78
				4353	**2.60**		
8. 1960 Ford Zodiac modified IL6 2.6L NA (Described in Sub Note A)							
752	79.4	2.381	3.0%	5600	1.836	1772	14.84
				4220	**2.57**		
9. 1983? US Racing Rolls-Royce Merlin 60V12 27L PC – Before fitting Dykes' rings							
753	152.4	2.381	1.56%	3800	1.693	1593	19.30
				3791	**2.41**		

It is probable that Hepworth used Nos 2, 4, and 5, *inter alia* in developing his relation, as described in the main text. Both PC and NA units appear in this data and the averages are:-

4060 mm.g **2.52 m/s**

These experimental results are plotted as **MPS** v. **√(S/w)** below.

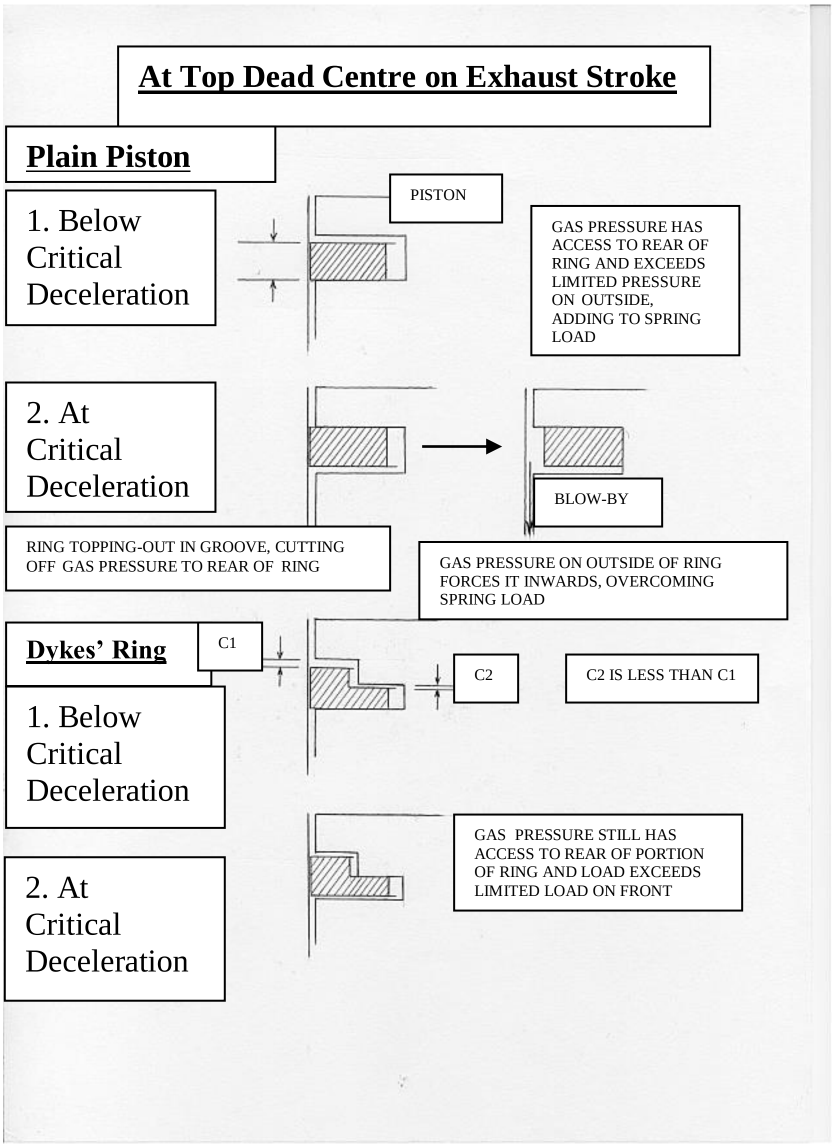


Sub Note C
Dykes' Rings

The first publication of Prof. Dykes' work was in 1947, with a final MIRA report in December 1951 (174). By that date the BRM 135V16 1.5L PC engine had already benefited from his invention. When using plain piston rings of 3/64 inch (1.191mm) width there had been serious blow-by at 10,000 RPM (limiting ring deceleration of nearly 107,000 ft/sec^2 (3318g)), **w.MPD** = 3952 mm.g (Example No 4 in Sub Note B). Fitting 2 x 3/64'' maximum width **L**- section rings (turned from plain rings) to each piston provided a cure up to the design speed of 12,000 RPM, equal to 4778g.

By 1956 the Vanwall IL4 2.5L NA was also using 2 x Dykes' compression rings per piston, and the 1959 –1960 Coventry Climax IL4 2.5L NA used them as the top ring. The same use was made of the type in the 1961 –designed 90V8 1.5L NA engines from BRM and Climax.

In the USA (for e.g.), the 1965 Ford 90V8 7L stock-car racing engine was fitted with Dykes' rings (220).



Postscript to Note 13 Part II
1954 Maserati 250F: Piston-Ring Flutter

Information on the Maserati 250F piston-ring width (1.5 mm in Moss' customer engine 2508 (882)) was received after the data in Sub-Note B was collected.

The 1954 250F engine, when run at the maximum speed of 7,200 RPM recommended to customers who did not wish to change bearings after every GP (147), was oil-tight (790). This represented **w.MPD = 4114 mm.g** and **MPS.√(w/S) =2.55 m/s**. The "works" cars were geared to use much higher RPM, beyond the power peak which was then 7,250 RPM (147), and, in equalling his 1951 Alfa Romeo 159 lap speed at Spa in June 1954 (at 120.5 MPH, 193.9 kph), Fangio used 8,100 RPM (and not his own car but the other "works" car of Marimon!). The engine threw out a great deal of its oil at this speed (790), which was 27% higher in **w.MPD** at **5207 mm.g**. In the race, presumably with the same engine, Marimon also used 8,100 RPM and had to stop for plugs after 1 lap and retired with terminal engine problems 2 laps later. Fangio won the race, with a fastest lap 1.3% slower than his practice speed i.e. not using the same RPM.

This incident does illustrate how piston engines could be run beyond the flutter boundary for short periods but with life-reducing consequences.

Maserati later improved their oil-cooling, by shifting the oil tank from the intake side of the engine compartment to the tail, connecting it with Al-alloy pipes (790)- which then caused their own race-losing fatigue failure at Monza later in 1954.

Precisely how the flutter problem was overcome is not known to this author.

Postscript 2 to Note 13 Part II
Confirmation of the hypothesis that 3NA engines at Peak Power run too fast for piston-rings to flutter

Ref (1062), published in 2004, gives the driver Martin Brundle's account of the McLaren-Peugeot fire on the grid for the 1994 British Grand Prix:
p.179 "*As the start lights came on, I held the engine revs as usual at 11,000 RPM. I looked in my mirror and saw a massive sheet of flame erupting from the rear of the car....the entire"* [engine] "*system had become pressurised and the engine had actually started to consume its own lubricant.....I was then blamed for using the wrong revs on the start line....It turned out that I was holding the revs in a zone in which* ***the harmonics made the piston rings leak*** *which ultimately allowed the pressure to go from the cylinders into the sump and send the oil on its way to create the pyrotechnics*". Peugeot had changed from 3 rings to 2. "*No-one had told me about that or* ***the crucial rev band***" [author's **bold**].

Peak Power RPM of the 1994 Peugeot was around 13,500 RPM. A guesstimate of the critical 11,000 RPM situation suggests that;

w.MPD was **4,400mm.g.**

Limiting Mean Valve Speed

DST 18 February 2003.

By analogy with the "Bottom-End" of a piston-engine it would be expected that the "Top-End" of a poppet-valve unit would be limited by "Mean Valve Speed", i.e.:-

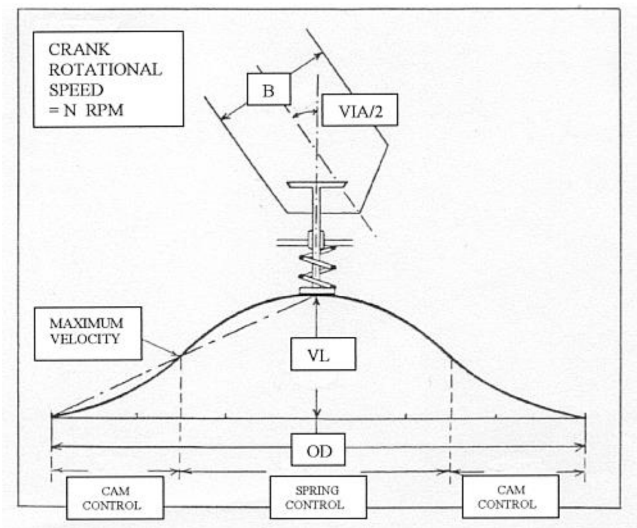
$$\left[\frac{2 \times \text{Valve Lift} \times \text{Crank RPM}}{\text{Valve Angular Opening Duration (Crank Degrees)}}\right] \propto \sqrt{\left[\frac{\text{Stress in Valve Springs}}{\text{Density of Valve Gear}}\right]}$$

With Valve Lift (VL) proportional to Valve Diameter (VD) and this proportional to Cylinder Bore (B) *for a given type of valve gear* (including in that a statement about the Included Angle between Valves (VIA), Number of Valves per Cylinder (VN) and also about Angular Opening Duration (OD), then a relation *at a given state of material technology* should be:-

$$B.N = \text{a Constant depending on type of valve gear}$$

However, despite the attraction of this broad "**Bore Speed**" analogy with Piston Speed, the details should be examined from first principles for valve gear. This follows in this Note , *for gear using return springs*. "Desmodromic" gear, giving positive control of valves at all times (within very tight tolerances), even when over-revved beyond specified limits, *should* be the answer to a racing-designer's prayer, but in practice only the Mercedes-Benz M196 of 1954-1955 (468) and Ducati in the motor-cycle sphere since 1956 and to date (74, 95, 762) have made a real success of this technology. Maserati (711), Norton (761) and Cosworth (59) are known to have experimented with it but not carried it through to racing. It may be that expense has played its part in this*.

In spring-controlled poppet valve-gear, the spring absorbs the energy given by the cam to the valve and all associated reciprocating or rocking parts during the first part of the opening, thereby bringing these parts to rest at maximum lift while they are just in contact with the cam (at valve-bounce speed) and with the spring at maximum designed stress. The spring then returns the gear by releasing this stored energy until reaction from the cam brings the valve to rest on its seat. This process applies to any type of spring, including an air or gas spring.
The process is sketched below in terms of Valve Lift v. Angular Opening. With a constant rotational speed the Opening corresponds to time, so that the slope of the curve represents valve velocity.



The Mean Valve Speed (MVS) is given by:-

$$MVS \propto \left[\frac{2 . VL . N}{OD}\right]$$

It is assumed that the cam characteristic is such that:-

Maximum Valve Speed $\propto$ MVS

Continued on Page 2

*OSCA raced an IL4 1.5L sports car with desmodromic valve-gear briefly in the 50s.

The *Maximum energy given to the Valve Gear*, whose total reciprocating mass is VGM (with appropriate geometrical adjustments to include the effect of any rocking parts and to include a proportion of the spring mass) is therefore:-

$$\propto \text{VGM} \times [\text{MVS}]^2$$

This energy has to be stored by compressing, and stressing the valve spring, additional to the pre- loading when the valve is on its seat. The pre-loaded energy can be taken as proportional to the additional energy (see Sub Note A).

The further analysis assumes a coil spring, which was by far the most-used type from 1906 to 1990. Part I of this Note includes data which shows that the steel coil springs of the Ford-Cosworth DFV were co-equal with the piston in limiting the engine's overhaul life.

For a relatively close-coiled spring, the torsional energy stored is given by:-

$$\frac{(\text{Torsional Stress})^2}{\text{Modulus of Rigidity}} \times \text{Volume of Spring} \qquad \text{Ref. (760)}$$

Therefore

$$\text{VGM} \times [\text{MVS}]^2 \propto \frac{(\text{Stress})^2}{\text{Modulus}} \times \text{Volume of Spring}$$

From which:-

$$[\text{MVS}]^2 \propto \left[\frac{(\text{Stress})^2}{\text{Density} \times \text{Modulus}}\right] \times \left[\frac{\text{Volume of Spring}}{\text{Equivalent Volume of Valve Gear}}\right]$$

the expression $\left[\frac{(\text{Stress})^2}{\text{Density} \times \text{Modulus}}\right]$ is the "**Valve-Gear Material Factor**" (**VMF**) (which assumes for simplicity that the spring and other gear are the same density) and, when it is at its limit according to the current state of material technology, it sets the limit to **MVS** according to the other factor $\left[\frac{\text{Volume of Spring}}{\text{Equivalent Volume of Valve Gear}}\right]$ which is the "**Valve-Gear Volume Ratio**" (**VVR**). This factor **VVR** will have characteristic values according to broad designs; for Double Overhead camshaft (DOHC) types, operating the valves through "inverted bucket" tappets, **VVR** will be high; for push-rod-and-rocker types with a low-mounted camshaft, **VVR** will be low.

$$\therefore \qquad \mathbf{MVS} \propto \sqrt{[\mathbf{VMF}]} \times \sqrt{[\mathbf{VVR}]}$$

MVS with known internal data

Where the data are known, MVS is therefore the guide to the "Top-End" limit of a specified type of engine at a given date, i.e. a given state of material technology. Usually the Inlet Valve is heavier than the Exhaust Valve (the Bugatti 3-Valve engines are exceptions) and so Inlet Valve Maximum Lift (**IVL**) and Angular Opening (**IOD** in crank degrees) are used to find **MVS**.

The Database uses:-

$$\mathbf{MVS} = \frac{\mathbf{IVLmm \times N\ crank\ RPM}}{\mathbf{83.333 \times IOD\ crank\ degrees}} \qquad \text{m/s}$$

Obviously, ways of raising N at a given value of the product of **VMF** and **VVR** are to reduce **IVL** by increasing the number of cylinders for the required Swept Volume and/or increase the number of valves per cylinder for the required inflow. Apart from the use of valve gear with the highest value of **VVR**, which can include, for examples, the rifle-drilling of valve-stems and making the valve heads hollow, a lower-density material may be used in the valve-gear. This latter was made possible by the development of a Titanium alloy (for US military aerospace purposes originally) which could withstand the exhaust valve temperature of a Normally-Aspirated (NA) engine for a sufficiently long racing life (there is no point in only using Ti-alloy for inlet valves, as the exhausts would then become the limiting factor). Ti-alloy was applied to Grand Prix engines from 1989, after the return to NA rules, although Honda had applied it to all valves in racing motor-cycles in 1983 (97) and big US V8 engines with push-rod gear had fitted all-Ti-alloy valves as early as 1962 (220).

B.N as a surrogate for MVS

If the required internal details are *not* available, the analysis may be continued as follows:-
where engines are operating to the limits of power and speed **IOD** will be similar for all engine types, around 320 to 340 crank degrees, so that the expression for MVS can be reduced to :-

$$\mathbf{IVL \times N} \propto \sqrt{[\mathbf{VMF}]} \times \sqrt{[\mathbf{VVR}]}$$

IVL is proportional to Inlet Valve Diameter (**IVD**) in order to pass the flow and **IVD** is related to **B** by **VNI** and **VIA**.
With 2 opposed, inclined valves per cylinder:-

$$\mathbf{IVL} \propto \mathbf{IVD} \propto \frac{\mathbf{B}}{\mathbf{2 \times \cos(VIA/2)}}$$

With in-line pairs of valves per cylinder (i.e. *not* radially disposed)**:-**

$$\mathbf{IVL} \propto \mathbf{IVD} \propto \frac{\mathbf{B}}{\mathbf{2}} \quad \textbf{irrespective of VIA.}$$

So the overall expression can be reduced further to:-

$$\mathbf{B.N} \propto \sqrt{}\mathbf{[VMF]} \times \sqrt{}\mathbf{[VVR} \times \mathbf{[\cos(VIA/2)\ or\ 1\ ]}$$

depending on whether a 2 or 4- valve-per-cylinder engine is being considered. It follows that, *at a given state of material technology*, if the required valve-gear life is restricting the operating speed, then:-

B.N = a Constant depending on type of valve-gear.

Alternative types of Valve Spring

The main text discusses, against the relevant designs, the use of Hairpin springs (which bend the wire) in place of Coil springs (which twist the wire). Also the introduction of air or gas springs (Pneumatic Valve Return System, PVRS) invented by J-P Boudy of Renault in 1984 (474), which superseded wire springs in all front-line Grand Prix engines in 1990.

Valve-Bounce Speed

The analysis above is derived from the requirement to restrict engine operating speed so as to achieve some specified life of the springs. The *engine* life, in a non-desmodromic-valve-gear high-compression unit, can always be cut to zero by over-revving until a valve leaves the cam (because the spring cannot absorb its energy within VL) and is struck by a piston. It is reasonable to assume that this "Valve-Bounce" speed is proportional to the time-limited speed.

Coil-Spring " Surge"

With coil springs the moving mass of the wire has its own natural frequency and higher rotational speeds may resonate with this (causing "Surge") and lead to premature fatigue failure. The analysis for this leads to an expression similar to that given above. It shows that the natural frequency may be increased by raising spring stress by increasing wire diameter and/or reducing the number of turns. This of course is limited as before by the state of spring material technology. Surge damping can also be introduced by an inner spring of different frequency having an interference fit against the outer coils, but this is of limited life before the fit wears out (e.g.,the technique was considered unsuitable for a 24-hour life for a Le Mans engine in 1993 (16)).

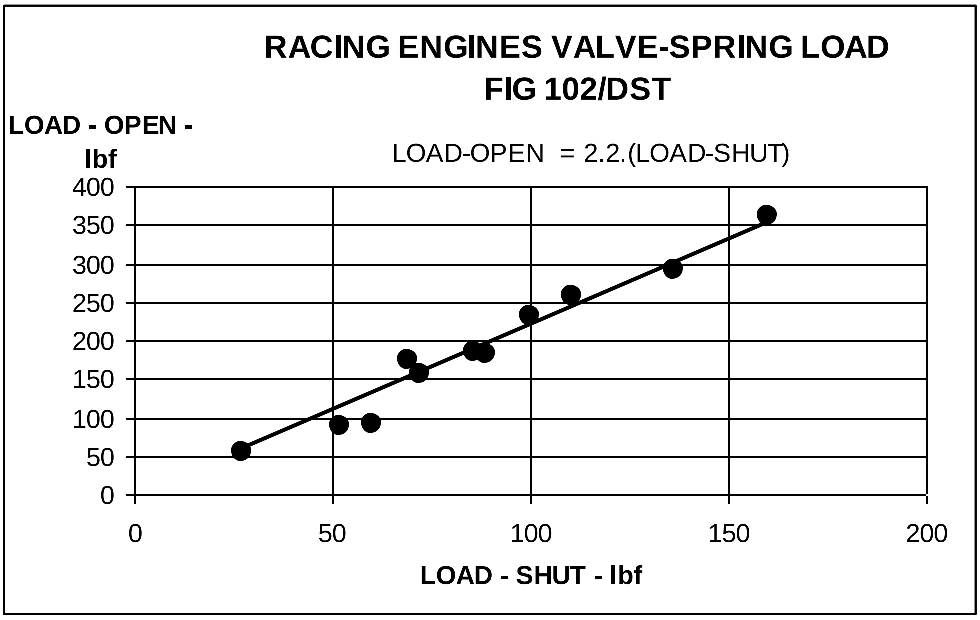
Hairpin and pneumatic springs do not suffer from surge.

See Sub Note A on following page

Sub Note A

Examination of valve-spring design details, in particular in Coventry Climax (33,34) and Ford-Cosworth (583) engines, over a range of valve diameters from 25 to 49 mm, has shown that the (Max. Open Load)/(Shut Load) ratio was very close to constant, at about 2.2 (see the details and chart 102/DST below). Therefore, with Energy Stored being proportional to $(\text{Load})^2$ (760), the (Additional Energy Stored)/(Pre-Loaded Energy) was around a constant value of 4 in a wide variety of engines.

VALVE-GEAR DATA No.	ENGINE	DASO	SPRING LOAD-SHUT lbf	SPRING LOAD-OPEN lbf
1	COSWORTH FVA	583	100	230
2	CLIMAX FWA2	33	60	92
3	CLIMAX FWA3	33	*72.3*	156
4	CLIMAX FPF 1.5	33	110.4	257.6
5	CLIMAX FPF 1.5 Mk2	33	136.5	290.7
6	CLIMAX FPF 2.5	33	110.4	257.6
7	CLIMAX FWM	33	27.2	55
8	CLIMAX FWMA	33	52	88.75
9	CLIMAX FWMV1 (Ass'd)	34	69	174
10	CLIMAX FWMV6 (Ass'd)	34	86	185
11	CLIMAX FWMW	34	89	181
12	DRAKE 2.8L TC	711	160	360



Note 13 Part 1

Sub-Note A addition 20 May 2019

Cosworth type TJ piston

In the web-site "f1technical.net/forum" the contributor "Mudflap" provided a detailed stress/temperature analysis of a Cosworth type TJ piston. This additional Sub-Note A Note 13 Part 1 shows the drawing of the piston from that source for comparison with the earlier pistons in that Note.

Cosworth TJ

Installed in 2003 JaguarR4 and 2004 Jaguar R5 (TJ perhaps indicates "**T**eam **J**aguar?).

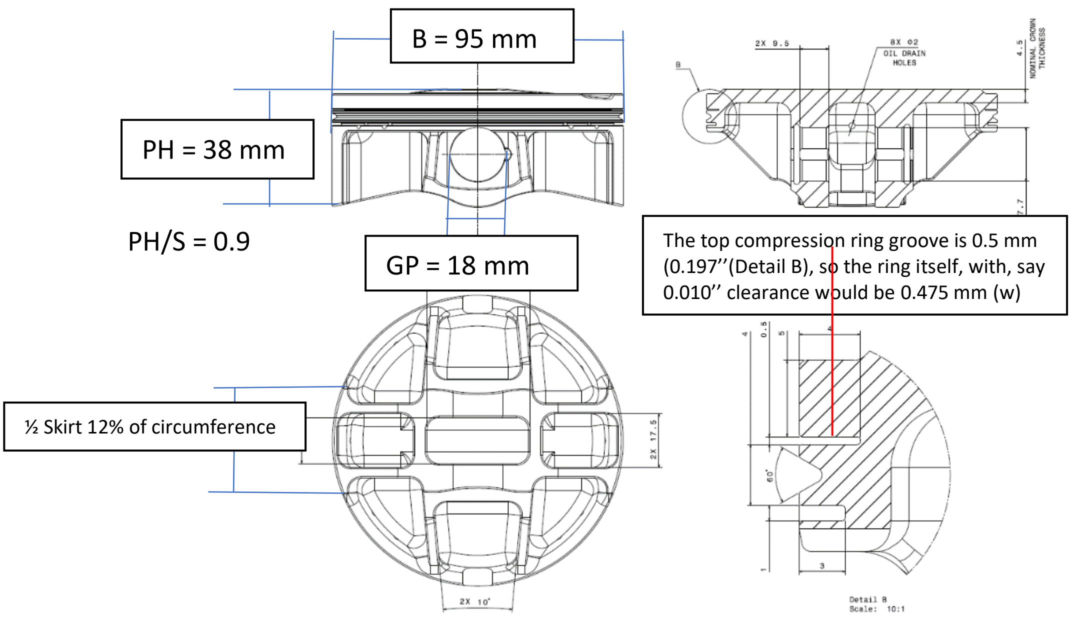
After Jaguar was sold to Red Bull, the TJ engine continued in their 2005 RB1

90V10 Bore (B) 95/Stroke (S) 42.3* mm =2.246; V = *Assumed 2998 cc.

Rev. limit 19,000 RPM. Assumed Peak Power (PP) speed (NP) was 18,500 RPM.

PP claimed 915 BHP.

BMPP = 14.76 Bar @ MPSP = 26.08 m/s



Gudgeon pin (GP)/S = 42%

Cut-away of skirt = 76%

Maximum Piston Deceleration at Peak Power RPM (MPDP), assuming that Con.-Rod Length (CRL)/Stroke (S) = 2.5, was 9,710g.

So, Top ring axial width (w) x MPDP = 0.475mm x 9,710g = 4,612 mm.g

Piston mass – not given.

This 2003 – 05 piston proportions are very similar to those given for 1996 pistons in Note 13 Part 1, **except** for the thinner top ring. The value of w.MPDP is not very far above the statistical critical value (flutter onset) of 4,000 mm.g given in that Note for rectangular iron rings. Density of the TJ ring is unknown, but may be lower than iron. Honda in 2006-2008 ran successfully with Ti-alloy (see Corrections & Additions, P.10, 30 October 2015), which was 60% lower density than iron.

Piston material

Given as 2618A. This is virtually the same as RR58, whose %age composition by weight (see Note 14) is:-

Si	Fe	Cu	Mg	Ni	Ti	Al
0.2	1	2.5	1.6	1	0.1	93.6 with control over various trace elements.

Cosworth developed their own process methods for RR58.

Note 14
Pistons for High-Power Engines

Materials (See the General Note on p6)
Cast-Iron and Steel
Effectively only cast-iron or steel were available for the pistons of the earliest internal combustion engines – the Lenoir gas engines from 1860, becoming petrol Otto-cycle engines from 1883 when Daimler built the first. With combustion temperature reaching over 2000C, even momentarily, it must have seemed obvious in any case that only such high melting-point materials (cast-iron 1230C; steel 1350C)(689) would be of any use. As late as 1913 a design offered for the Kaiserpreiz aero-engine competition with aluminium-alloy pistons was rejected simply on the grounds that such material could not be strong enough since the melting-point of aluminium was only 659C(468).

Cost then dictated cast-iron originally and this was universal in racing until 1906. Cast-iron Grand Prix (GP) engine pistons, e.g. Renault type AK 12.8L, In-Line (IL) 4cyl 165mm bore x 150mm stroke (henceforth Configuration x B x S in that order, figures only) meant a Mean Piston Speed (MPS) around 6 m/s at that date. It was equally obvious that piston mass must be minimised if MPS and therefore power was to be maximised. It is believed that Frederick Lanchester first used machined steel pistons in 1905 for a touring car (2). An advantage in weight would be secured in that 0.2%C/Fe mild steel permitted tensile stress 4 x the cast-iron limit (at room temperature) and so sections could be machined thinner. Maurice Sizaire certainly adopted steel pistons machined from solid billets in his pioneering and successful long-stroke 1907 Coupe de l'Auto 1.2L engine (1 cyl. x 100 x 150), to reach MPS = 12 m/s (361). Kept secret for 57 years afterwards, until the designer revealed it, was that these pistons had been machined too thinly and began to fail in the race. Clearly this had been anticipated and spares provided and, although replacement was against the rules, the drivers went off-course into a wood near the long circuit and changed the parts unobserved by stewards – not too lengthy a job with 1 cylinder (361). The lap time increases were put down to "calls of nature"! The 1907 Sizaire et Naudin was not at the limit of steel, however, - at least for a *brief* burst of speed – because the following year the stroke was lengthened to 250 to give 2L and MPS = 20 m/s (259). It appears that the 1906-07-08 S & N engines were actually limited by mean valve speed, not mean piston speed (the surrogate valve-speed factor "Bore x RPM" being constant at 4 m/s).

Other racing engines then followed the Sizaire lead to steel pistons (and low B/S ratio) and they remained the usual GP practice until WW1, although the 1914 French GP-winning Mercedes type M93654 4.5L (IL4 x 93 x 165) still used cast-iron. It reached MPS = 17 m/s (468) so that the castings must have been of improved quality since 1906.

Aluminium alloys pre-World War 1
The search for lighter pistons (Note I) had already led some designers to try aluminium regardless of its low melting-point and reduced hot strength, the attraction being an elemental density of 2.7 g/cc, 66% below the 7.87 g/cc of iron (alloying alters both figures). The discovery by Wilm in 1909 of age-hardening to give higher strength, after quenching from high temperature, when applied to an alloy comprising 4.5%Cu/0.5%Mg/0.5%Mn/94.5%Al (hereafter figures only), which he named "Duralumin" after his employer Durener Metallwerke, describing its properties in 1911 (667), opened-up new possibilities. Cappa of Aquila-Italiana used Al-alloy pistons, composition unknown but possibly Duralumin, in a 4.2L (IL6 x 82 x 132) non-GP racing engine in 1912 (621) which reached MPS = 16 m/s. Although one source (687) credits Cappa with using Al-alloy pistons in 1906 it seems doubtful as being pre-Wilm so that the metallurgy would have been suspect. The same source gives the French Corbin foundry as supplying Chenard-Walcker and Panhard with Al-alloy pistons before 1914. These would have been touring-cars. Source (663) says that the 1914 GP 4.5L Peugeots (IL4 x 92 x 169) had Al-alloy pistons. It is also known (617,413)* that Rolls-Royce tried them in the developed "40/50" 7.4L (IL6 x 114.3 x 120.65) engines built for the 1913 Alpine Trial, where power was wanted regardless of "cold-piston-slap" (due to the clearance necessary to accept the x2 expansion coefficient of Al compared to the cast-iron cylinder) (Note II). (**But see a correction in Note VIII*).

Refs (6) and (687) state that Miller in the USA produced Al-alloy pistons in late 1913 in a formula of his own which he named "Alloyanum" including Ni as well as Cu – which anticipated UK research (see later). Pistons of this sort were fitted by Miller in a 1914 GP Peugeot which finished 2^{nd} in the 1915 500 mile Indianapolis race and won in 1916 (687), and were almost certainly in the same or a sister car which won in 1919.

Daimler tested Al-alloy pistons (probably Duralumin) in its Mercedes 1914 GP engine satisfactorily but, given the choice of those or cast-iron, the works drivers (who were also experienced manufacturing men) preferred the low-risk of the latter, as already mentioned (468). The result of a 1^{st}, 2^{nd} and 3^{rd} against strong opposition over 752 km justified their decision!

More-or-less at the same date as the Miller and Daimler work, Walter Bentley in conjunction with his car suppliers DFP in France had obtained Corbin-cast pistons with the specification 12Cu/88Al and used them to raise the power of

the 4 cyl. 2L DFP engine for class record-breaking at Brooklands (688). Although post-Wilm this alloy owed nothing to his work and was not heat-treated (667 – which has an incorrect chronology as pre-Wilm). These Corbin pistons overcame the piston failures which had previously limited power increases from the DFP, where cast-iron had cracked and very-light steel had broken their rings (688).

These early users of Al-alloys discovered that the low melting-point was irrelevant because the high thermal conductivity (4.5 x cast-iron) dispersed the heat too efficiently to affect the metal, provided that the crown and top skirt sections were adequate to pass this heat to the rings and hence to the cooled cylinder wall, or to oil splash under the piston. Tests later (671) would show that the crown temperature was actually around 200C lower than cast-iron and this produced the serendipititious gain that compression ratios could be raised on a given fuel without knocking. Bentley certainly took advantage of this in his 1914 DFP (688). A 1930 quotation can be inserted here to summarise qualitatively the gain from Al-alloy pistons:- "Light weight gives improved accelerating properties and the high thermal conductivity with absence of hot-spots gives good high-compression performance" (677).

Al-alloys in WW1 Engines

After WW1 broke out Bentley was sent by Briggs of the Royal Navy to Rolls-Royce and Sunbeam, who were designing new aero-engines, to show them his DFP results with Corbin piston alloy (688). The former company had tested their prototype 20.3L (V12 x 114.3 x 165.1) design with cast-iron pistons at 4.5 compression ratio in February 1915 and found that these burnt through the crown in only 20 minutes (617).* They were replaced successfully by Corbin-alloy parts with the crown thickness graduated according to the area receiving heat and skirts of the "Zephyr" type in which the middle third of the piston did not touch the wall (617,688). Sunbeam also adopted Corbin-alloy in their later WW1 aero-engines, which were based on pre-War Peugeot technology, and also used it in a racing engine for the first time in the 4.9L car (IL6 x 81.5 x 156) built surreptitiously for the 1916 Indy 500 in neutral USA (24). It finished 4th. (* *But see a correction in Note VIII*).

The Corbin-alloy, given the official designation L8 (645), was used by Ricardo for gravity die-cast pistons in his 1917 18.3L tank engine (IL6 x 142.9 x 190.5) (242) (which made the Mk V tank a really effective weapon) and in Wolseley-built Hispano-Suiza 11.8L aero-engines (V8 x 120 x 130) (671). The French-built units also, of course, had Al-alloy pistons and presumably of the same alloy.

WW1 Research on Al-alloys

Much UK national research effort was put into Al-alloys during WW1 at the Royal Aircraft Factory (RAF – later renamed the Royal Aircraft Establishment, RAE, after the 1 April 1918 formation of the Royal Air Force) and at the National Physical Laboratory (NPL). Test data were published after the War (671). Particularly interesting comparisons were made in a 1-cyl. portion of the RAF type 4d 13.2L aero-engine (V12 x100 x 140), which had 2 inclined overhead valves in an aluminium head and was air-cooled. The results were:-

1 cyl. 100 x 140 = 1100cc: all fully-skirted pistons: PH/B = 0.81: PH/S = 0.58

Piston ref.*	K	G	G	D
		Early-standard		1918 Standard
Piston Material	Cast-Iron	B4 alloy:- 7Cu/1Sn/1Zn/91Al	B4	B4
Bare Piston Mass g	806	573		647
Piston Assembly Mass** g	1113***	880		954
	Datum	**-21%**		**-14%**
Compression Ratio R	4.6	4.6	5.3	4.6
Mean Piston Speed (MPS) m/s	8.4	8.4	9.3	8.4
HP/Litre	15.5	16.4	17.7	16.8
	Datum	**+6%**	**+14%**	**+8%**

* Arbitrary report designations.
** Including estimated mass of 2 iron rings and 23mm o.d. steel gudgeon pin.
***Adjusting from: 3 rings and 18mm o.d. pin: to 2 rings and 23mm o.d.pin as used for the Al-alloy pistons, changes which were not relevant to the material.

In these figures note the smallness of the weight reduction between cast-iron and Al-alloy after the necessary thickening-up of the metal sections with the lighter material because of its lower hot strength, plus the unchanged rings and gudgeon pin which have to be added to the reciprocating mass. The extra mass of the 1918 standard, Piston D, which had much under-crown ribbing, indicates that the first Al-alloy design was not strong enough for the desired life – which would have been only 100 hours at most. The report (671) did not identify the reasons for the 6% to 8% power gain between Pistons K and G or D, running at the same speed, but it is deduced that these were:-

(1). Reduced friction from the lower side-thrust of the lighter piston;
(2). Ignition advance possible without knocking because of the 200C cooler crown.

Piston G also had a 4% better Specific Fuel Consumption than K. The heavier Piston D compared with G ran 10C cooler, presumably because the ribbing assisted heat transfer to oil splash, and could therefore have had more ignition advance and power. Reason (2) may therefore be more significant than (1).

The RAF took further advantage of a somewhat lighter and much cooler piston than K to raise both speed and compression ratio,as listed in the 3rd column, and gained another 8% of power. What the effect on life would have been cannot be known.

Racing Engine Material Change Comparison Post-War

The first Post-WW1 French Grand Prix, in 1921, was won by a Duesenberg 3L (IL8 x 63.5 x 117.475) from the USA, which very likely was fitted with Miller Alloyanum pistons. At 16.6 m/s its MPS showed no advance on the 1914 cast-iron Mercedes.

However, an interesting practical example of piston-material change, in a racing engine, is recorded in (645). A Peugeot 1913 GP 5.7L (IL4 x 100 x 180) – the actual winner of the French GP of that year – was brought to Brooklands post-WW1 and, over 1922-24, was fitted with various Al-alloy pistons (amongst other modifications). Those which were effective represented, for the reciprocating assembly, a mass reduction of about 40% from the original steel-piston assembly. Pistons representing assembly reductions of 60% and 50% suffered failures at very short lives. One reason for the much greater mass reduction in this Peugeot, compared with the RAF 4d given above, would be that the racing distances required from a set of pistons were so short compared to the 911 km of the 1913 GP – although admittedly flat-out on the Brooklands Outer Circuit with MPS up to 14.8 m/s, the pistons were changed after each season, at most. Another reason for the extra weight reduction may have been that the post-War pistons were of lower PH/B ratio than the 1913 parts, probably as 1 to 1.3 (597). The lower PH/B had been used in the 1915 Peugeot V8 aero-engine which was based on two blocks of the 1913 racing design (400,371).

Improved Al-alloys

The Al-alloy research at NPL under Rosenhain in WW1 led (amongst other useful materials) to "Y-Alloy" for pistons, comprising 4 Cu/2Ni/1.5Mg/92.5Al, which retained its strength better than Duralumin at piston temperatures (665), the figures being: (from 678, a 1933 paper)-

		At 250C		
		Duralumin	**Y-Alloy**	
Max. Tensile Stress	**Tons/Sq.In.**	**17**	**21**	**(+24%)**
	MPa	**262**	**324**	

Unfortunately a comparison with "Corbin alloy" was not given in the ref. paper. It is a curious fact that historical review studies written 70 years later, egs. 665, 666, 667, do not mention either Corbin or Bentley. Their 12Cu/88Al alloy, which performed such outstanding service in WW1 and continued in touring-car engines until the late '20s (698,699), receives only passing mention (out of chronological order) in (667). It is true, of course, that this high-Cu variety did not father any later material developments, unlike the "professional" NPL research.

Magnesium alloys

Having succeeded with Al-alloys some designers thoughts post-WW1 turned naturally to Mg (density 1.74 g/cc, a 36% reduction from Al; melting-point 649C). Ref. (468) records that a composition 13.5Cu/0.5unknown/86Mg, which won a piston-design competition in Berlin in 1921, was used for forged pistons in up-rated 1914-type 4.5L Mercedes GP engines which at least finished in the 1922 431 km Targa Florio, although in lowly positions. This alloy is not known to have been used in later Mercedes designs but presumably was unsuited to the Pressure-Charged (PC) engines which were built shortly afterwards.

Villiers tried Mg pistons in Mays' highly-tuned Bugatti 1.5L (IL4 x 69 x 100) in 1923 but their crowns were nearly burnt through after little running (446).

To complete the mention of Mg for pistons at this point, since it seems to have been a dead end; Taylor used it satisfactorily in his first Alta Normally-Aspirated (NA) 1.1L (IL4 x 60 x 95) of 1928 (his later pre-WW2 engines were PC and employed Al-alloy pistons; Mahle die-pressed some Mg-alloy pistons for the special BMW type 328 2L (IL6 x 66 x 96) engines which took 1st and 2nd places in the 1940 closed-circuit Mille Miglia. In this case, the crowns were Cr-plated to prevent burning (30,695). Again, nothing seems to have followed from this. (Later: see Note IV).

Al-alloy Mainstream

Returning to the aluminium mainstream, pistons in Y-Alloy post-WW1 were cast and the material was not easy to forge. The Peter Hooker company were asked by NPL to work on this latter aspect in the mid-'20s and, when Hooker's went bankrupt in 1928, one of their staff, Devereux, formed High Duty Alloys (HDA) and carried on the research (666). Some of this was in conjunction with the Rolls-Royce laboratory and HDA were permitted to identify their "Hiduminium" products with the suffix "RR". The forging work by HDA led to Hiduminium-RR56, formula 2Cu/1Ni/0.8Fe/0.8Mg/0.3Si/0.1Ti/95Al (703), which was used extensively for pistons in the UK in the '30s (666). The 0.1% Ti content was important as it refined the grain to improve the fatigue resistance. (Later: see Note V).

Ref. (666) records that HDA, after Devereux's persuasion of the Italian Marshal Balbo, provided forged RR56 pistons very quickly to replace the sand-cast parts in the twin Isotta-Fraschini engines of Balbo's Savoia-Marchetti S55 flying-boat after he landed in the UK in July 1933 and before he led his mass formation of these aircraft across the Atlantic (Note III).

However, RR56 was not recommended by either HDA or Rolls-Royce for the higher temperatures which were applicable to PC engines (678) and by 1933 it was superseded in such applications by Hiduminium-RR59 :- 2.5Cu/1.6Mg/1Ni/1Fe/1Si/0.1Ti/92.8Al (618,666,667,703) which compared in properties with Y-Alloy as follows (678);-

		After 30 min. soaking at 300C		
		Y-Alloy	**RR59**	
Max. Tensile Stress	**Tons/Sq.In.**	**13.7**	**15.5**	**(+13%)**
	MPa	**212**	**239**	

Forged RR59 pistons were the mainstay of UK high-power reciprocating aero and some racing engines up until at least 1966, eg. used in the Cosworth type FVA 1.6L (IL4 x 85.72 x 69.14) Formula 2 unit of that year, which began a new age of design (583). An example of piston material change-over to this alloy is given by the racing motor-cycle engines built by Velocette in 1939: their long-developed 350cc (1cyl. air-cooled x 74 x 81) NA machine had sand-cast Y-Alloy (73); their new supercharged 500cc (2cyl. contra-rotating air-cooled x 68 x 68.5) design (the "Roarer") had RR59 (652). On the other hand the 1st post-WW2 UK racing engine, the BRM T15 1.5L (V16 x 49.53 x 48.26) designed in 1948 originally for 2.8 atmospheres absolute inlet pressure (448) used forged Y-Alloy pistons at least up to 1951 (693). The small bore probably eased the cooling problem.

Hiduminium-RR58

HDA developed in WW2 for the Whittle jet engine centrifugal compressor impeller a variant of RR59 which could accept higher stresses at about 250C. This was done by reducing the Si content and adopting a new production process – semi-continuous casting (666). The formula was:- 2.5Cu/1.6Mg/1Ni/1Fe/0.2Si/0.1Ti/93.6Al (670, 703). This was designated Hiduminium-RR58. It was produced for many other medium temperature applications subsequently, including the structure of the Anglo-French "Concorde" Mach 2 aircraft (selected in 1962). It is not known which racing engine first used it for pistons, but it was in the Coventry-Climax FWMV Mk5 1.5L (V8 x 72.39 x 45.47) by 1965 at latest (34).

The RR58 alloy was still used in front-rank GP engines in 1996. A case is known where repeated piston failures in this material in a 3L (V10) running to about MPS = 25 m/s were cured by outside computerised assistance which analysed the stresses transverse to the forging grain more accurately and redesigned to reduce them. The unit concerned has been very successful subsequently (561). (Later: see Note VI re Thermal Barrier coatings).

Beryllium-Aluminium Alloy

A step forward was taken in 1998-2000 when Be-Al alloy pistons were used in the Mercedes-Benz-Ilmor types FO110G, H and J 3L (V10)(700) (the material was also employed for cylinder liners). These pistons were tried first in very-short-life high-power Qualification engines, and then, when proved, in races. The material could have been "Lockalloy", 62Be/38Al (692), developed originally for spacecraft structure in the '60s and later made available commercially. The elemental advantage of Be can be seen as follows (689,692):-

		Al	Be	
Density	**g/cc**	**2.70**	**1.83**	**(-32%)**
E : Modulus of Elasticity	**Mpsi**	**10**	**42**	**(x 4.2)**
(At room temperature)	**GPa**	**69**	**290**	
Melting-Point	**C**	**659**	**1281**	

The 62/38 alloying with Al produces a Tensile Strength/Density ratio equal to the best Al-alloy, at 210 MPa/(g/cc) but with an E/Density ratio 3 x higher, at 72 GPa/(g/cc) (696), both at room temperature. To judge by the melting-point advantage these relative gains would be bettered at high temperatures.

However, the cost of this Be/Al alloy when it was first introduced for GP disc brake calipers in 1996 was 6 x Al-alloy (419), which was regarded as excessive even by Formula One standards. Consequently its use in calipers was banned from the start of 1998, and further thoughts by the regulatory authorities led to a general ban on any metallic material in the engine with an E/Density ratio over 40 GPa/(g/cc) from the start of 2001 (700). Illien of Ilmor has since stated that the more expensive Be/Al parts had actually lasted longer, which offset partly the first cost.

A new Al-alloy has been developed to take the place of Be/Al and this is said to provide a small gain over RR58, although not having the weight and heat rejection advantages of the banned material (700). (Later: see Note VII).

SUMMARY of MATERIALS and PROCESSES

To give an overview of the piston alloy changes through the years, the following table consolidates the figures:-

Average %ages

(Excluding controlled %ages of impurities, such as S and P)

Name	Date													Data Source
		Cu	Mg	Mn	Ni	Si	Ti	Sn	Zn	Al	C	Fe	Be	
Cast-Iron	Up to 1914					3					4	**93**		618
Mild Steel	1907-14										0.2	**99.8**		618
Duralumin	1909	4.5	0.5	0.5						**94.5**				667
Corbin (L8)	1913	12		*						**88**				688
Alloyanum	1913	?			?					**?**				6,687
RAF B4	1917	7						1	1	**91**				671
Y-Alloy	1919	4	1.5		2					**92.5**				665
Mg-alloy	1921	13.5	**86****											468
RR56	Ca. 1930	2	0.8		1	0.3	0.1			**95**		0.8		703
RR59	Ca. 1933	2.5	1.6		1	1	0.1			**92.8**		1		618,666,667,703
RR58	1943.	2.5	1.6		1	0.2	0.1			**93.6**		1		670,703
Lockalloy	Ca. 1965									38			**62**	692
New Al-alloy	2001									**?**				700

*May have contained a small amount of Mn (698).

**0.5% unknown.

A few words are appropriate to summarise the racing piston manufacturing processes. Iron items were cast, of course. Steel pistons were machined out of solid forged billets. Al-alloy parts began as sand-castings. In 1926 die-casting was used by Delage, for a lighter piston at higher cost for a few racing units (gravity die-casting had been used for quantity production of the Ricardo tank engine in 1917 (242), and mass-production used pressure die-casting from 1927 (668)). However, sand-casting remained in use by some engine-makers up to 1956, although forgings were available from the early '30s. The difference this better process could make was illustrated dramatically in the Connaught-Alta 2.5L (IL4 x 93.5 x 90) engine in mid-1956; whereas their sand-cast pistons would last a GP at MPS = 20 m/s, they would fail in fatigue at a short life when run at 21 m/s (+4.5%), which was needed for a competitive performance. The forged pistons which were then fitted were safe up to 22.5 m/s (+12%(701).

When Be/Al pistons were used in 1998-2000 by Mercedes-Benz-Ilmor, they were machined (with some difficulty and with special precautions taken against toxic dust) from the solid (700), probably from extruded blocks as had been the method with Be/Al brake calipers in 1996-97 (707). It was reported that castings were available later, although not used by Ilmor (700).

Some thoughts on the Origins of Piston Material changes

Reviewing the origins of piston material changes in high-power engines, the first major move, from cast-iron to steel and high MPS, was begun in 1907 by Maurice Sizaire. He had been trained as a builders' draughtsman and therefore was uninhibited by then-current engineering ideas.

The change from Fe-based to Al-based alloys, considered over the whole of the 90 years following 1911, was fathered by Wilm's research leading to Duralumin. His work was aimed at a military purpose (the manufacture of lightweight cartridge cases (667) – which did not happen). However, the 1913 Corbin 12%Cu/88%Al-alloy, un-heat-treated, appears to have been quite independent of Wilm's age-hardening discovery. The Corbin alloy was used first in car engines and only later for military purposes in the vast WW1 production of aero- and tank-engines. Miller in the USA also did his own light-alloy research in 1913 and made "Alloyanum" pistons for others before he built his own engines.

During and after WW1 and during WW2 all the Al-alloy improvments, starting from Wilm's basis, were aimed at military engines and were adopted later for cars.

The time-spans between a new alloy formula being proved and the last-known choice for a racing engine of the previous specification are interesting:-

- Y-Alloy last chosen after RR59 introduced:- 1948 – 1933 = 15 years;
- RR59 last chosen after RR58 introduced:- 1966 – 1943 = 23 years.

If Lockalloy *was* the Be/Al alloy used in 1998-99, this was some 30 years after it was available commercially. These long time-spans suggest a conservative approach even in competition engine design. The fact that all new alloys since 1919 (at least until 2001) have originated from Government-funded research for military ends reflects the very high cost of bringing into service a new high-duty material. Even when a high level of failure can be accepted as the price for a winning advantage on the majority of races, the R & D cost of developing for themselves an improved alloy seems to have been beyond even the well-funded Grand Prix engine makers of the last decade.

The 2001 regulation change banning materials of Stiffness/Density ratio greater than 40 GPa/(g/cc) might well have ossified the situation, but it actually seems to have stimulated an improved Al-alloy at last, of unknown origin.

Postscript

It is regretted that this description of piston alloy development has concentrated on UK and some US experience. The excuse is the same as Dr. Johnson's, when reproached for a dictionary error – "Pure ignorance!" – in this author's case of foreign languages.

Notes

General

Wherever possible this paper quotes the average major chemical constituents of engine materials, since this is fundamental data. However, it must be remembered that in practice:-

- Specifications give acceptable %age ranges for each major element so as to reduce preparation cost – just as machined parts have tolerances;
- Specifications also give maximum %ages of impurities such as S and P because keeping these under control is essential to achievement of the declared properties;
- The manufacturing processes for the material also have to be developed and controlled – these have been improved over the years to give better properties even with the same chemical composition;
- The heat treatment of material is a vital part of the specification.

These details have had to be omitted, partly through lack of information and partly as being not appropriate for a general review.

Note I. Louis Coatalen, long-time Chief Engineer and Managing Director of Sunbeam, had a dictum in 4-cylinder pre-balanced crank days (which he undoubtedly did not mean to be taken precisely) that "An ounce off the piston is worth a pound off the crankshaft and a hundredweight off the chassis!" (24).

Note II. The problem of "cold-piston-slap", which made Al-alloy parts originally a complete non-starter for the normal Rolls-Royce luxury cars, was solved in 1919 by Hives, their experimental chief. He provided close-clearance Al-alloy pistons with axial slits in the skirts so as to take up the relative expansion (617).
It is reported that Harry Rush and Riley (Coventry) Ltd gained a joint Patent, No. 139,351 dated 22 April 1919, for an Al-alloy piston with 4 axial slots in part of the skirt for the same purpose (1023). How Hives' design stands in chronological or legal relation to this is unknown.

Note III. The Balbo story does not relate whether all 48 engines of the formation were modified or only the Marshal's – and , if the latter – what the other 23 aircrews thought about it! Nor does it explain how the Modification Procedure was handled – "Standardisation Without Test" presumably. Anyhow, it succeeded!

Notes added 15 October 2002/ 11 April 2005.

Note IV. Magnesium alloy pistons. Ref (468) states that the 1923 Benz RH 2L Grand Prix engine (IL6 x 65 x 100) had Mg-alloy pistons produced by Hellmuth Hirth. The engine did not give sufficient power to be competitive (although the car was well ahead of its time by being mid-engined).
Ref (740) states that the 1994 Cosworth Ford Zetec-R 3.5L Grand Prix engine (75^0V8 x 100 x 55.69) had pistons of forged Mg alloy*. It powered the 1994 Drivers' Champion. By this date it was standard GP practice to rebuild engines after use in one Qualification session (4 separate flat-out laps), fit another less-highly-stressed engine for the 300km race and then rebuild that also.

Note V. High Duty Alloys and Rolls-Royce. This author is uncertain of the precise origin of some of the Hiduminium-RR Al-alloys referred to in the text. Ref (666) by W.Doyle, former Technical Director of HDA, states "All the development of *forged* alloys was carried out by HDA" (this author's italics). Ref (718), seen later, ascribes to Hall and Bradbury of the Rolls-Royce Laboratories not only the casting alloys RR50 and RR53 but also the forging alloys RR56 and RR59 and adds "HDA was chosen….to produce them…".Ref (718) also notes that when RR58 was Patented post-WW2 it was in the names of two metallurgists, one from R-R and one from HDA.

Note VI. Thermal Barrier Coating for piston crowns. Such coatings are now available to reduce heat flow into the piston. Ref (1051) reports that the ceramic coating "Keronite", new for 2002, had been shown by tests to reduce under-piston temperature by 60C.

Note VII. Jurgen Hubbert, racing manager for Mercedes-Benz, admitted at the end of the 2001 season that the loss of Be/Al-alloy for pistons had a bigger negative effect than had been expected. This suggests that the new Al-alloy had not performed as forecast.

Note VIII: see overleaf.
(*But see a correction in Note IX overleaf).

Introduction of Al-alloy pistons to Rolls-Royce engines

Research during 2005 in the archives of the Sir Henry Royce Memorial Foundation (SHRMF) at Paulerspury and of the Rolls-Royce Heritage Trust (RRHT) at Derby has established accurately how Al-alloy pistons were introduced into Rolls-Royce car and aero engines, in order to increase volume-specific power output from increased piston speeds and compression ratios.

Firstly, to correct a misstatement in the main text of the Note, the 1913 Alpine Eagle 40/50 engines did *not* incorporate Al-alloy pistons instead of Cast-Iron. The Engine Parts Register ('E' code) exists in the RRHT archives, from E1 dated 17 Sept 1909 onward. This Register records the material of the piece listed. It has been searched by Mike Evans and no Al-alloy piston is listed for the period covering the 1913 Alpine Rally engine manufacture (his research was described in Rolls-Royce Enthusiasts Club (RREC) Bulletin No. 271, July/August 2005).

Secondly, it is necessary to correct another misstatement that the Cast-Iron pistons of the first Rolls-Royce aero engine (the V12 later named "Eagle") were burning through the crown in short order during early tests beginning in late February 1915.This information was provided by Maurice Olley some 50 years later (617) but it is not supported in any way by the letters of Henry Royce at the time from St Margaret's Bay (where Olley was also based at that date) reacting to the test reports being received from Derby (ref. 305, the "*Blue Book*"). Not only is there no mention of anything so serious as holes in the pistons but the engine designed for 200HP @ 1,600RPM was within 4 days tested at 225HP at that speed, had done a 4 hour continuous run inside 8 days and in 26 days had completed 6 hours continuously at 2,000RPM (+ 56% stress over the design RPM). These results could not have been obtained if there had been any significant short life trouble with the Cast-Iron pistons. It is certain that Olley was confused in his dating with a piston failure nearly 3 months into the bench development programme when running deliberately with weak mixture and probably at substantial parts lives. This failure was discussed in a 19 May 1915 letter, with Royce writing that it was "*quite evident that the centres of these*" (original type of) "*pistons are too hot*". He mentioned further on 9 June 1915 that, before the weak mixture test the pistons had "*behaved exceedingly well during normal running*". Many different types of Cast-Iron piston had been considered and continued to be designed and tested.

When W.O.Bentley gave his original (1958) reminiscences of visiting Derby to see Ernest Hives, the man in charge of the V12 testing, with his DFP Corbin-Al-alloy piston, his description suggested that it was within a few months of the outbreak of war on 4 August 1914. It is now known, because of a discovery in the SHRMF archive by the late Peter Baines (as recorded by Mike Evans in RREC Bulletin 270, May/June 2005) that it was actually not until a day or so before 8 July 1915. On that date Hives wrote to Royce to describe Bentley's visit and his handing over of the piston. Mike Evans has shown(Bulletin 271) that the consequence was a 1st all-Al-alloy* piston, part number E6806, dated 29 July 1915, actually for the standard 40/50car engine. Evidently this was for a trial test on a well-understood engine. An all-Al-alloy* piston for the V12 aero engine followed, E7031 dated 20 August 1915.

Rolls-Royce used the Corbin alloy for their Al-alloy pistons, i.e. 12% Cu, 88% Al (afterwards coded officially L8) for some 2 years but a letter of Royce to Harvey-Bailey (senior) at Derby, dated 8 August 1917, stated that the Air Board (of the Ministry of Munitions) had instructed a change to the (Royal Aircraft Factory (R.A.F.)/National Physical Laboratory) alloy B4. This had 7% Cu, 1% Zn, 1% Sn, 91% Al. Royce stated "*This matter is most urgent, as I consider the present alloy a source of danger*". Standardisation of B4 for pistons was therefore effected by X3136 dated 28 August 1917, to go into Eagle Series 8 from the imminent start of production of that development (which provided 350HP @ 1,800RPM, i.e. a 75% increase over the original contracted power). A description of various tests of B4 by the R.A.F. is included in the main text. It appears, from lecture notes on materials produced by Harvey-Bailey 20-odd years later (in 1939) that the problem with the Corbin alloy was that, with 12% Cu, it continued to age in service and become brittle. The B4 alloy, still being high in Cu, would have had the same defect but to a lesser degree.

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*Royce had had schemed and tested *composite* pistons after the Bentley information was given to him, having Al-alloy crown but Fe body, between 12 July1915 and 22 September 1915 but, at the latter date, decided that the all-Al-alloy piston was best (305).

Note IX

Cosworth type EC, aka Ford Zetec-R

The author was told in 2011 by Martin Walters, who developed this engine, that the Mg-alloy piston was not raced, having too short a life. The EC pistons were RR58 produced by a Cosworth-improved forging process.

NB. Correction Notes added in 2012

Because misstatements in the original text were from respected sources which are still extant and might be accessed they have been left in place and Notes VIII and IX added to draw specific attention to the earlier errors.

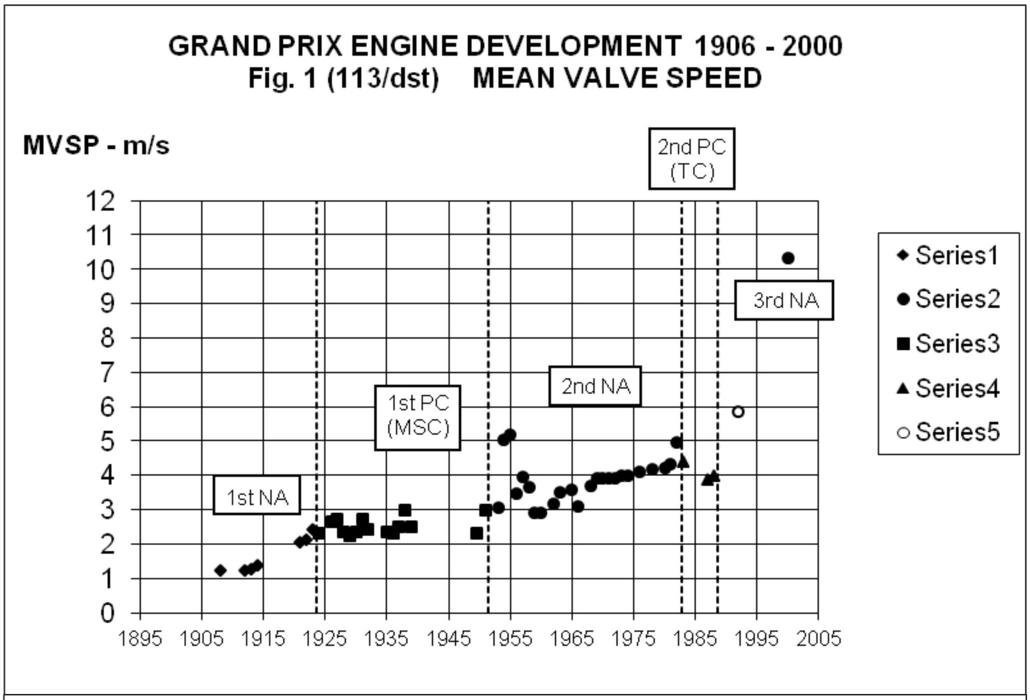
Note 15

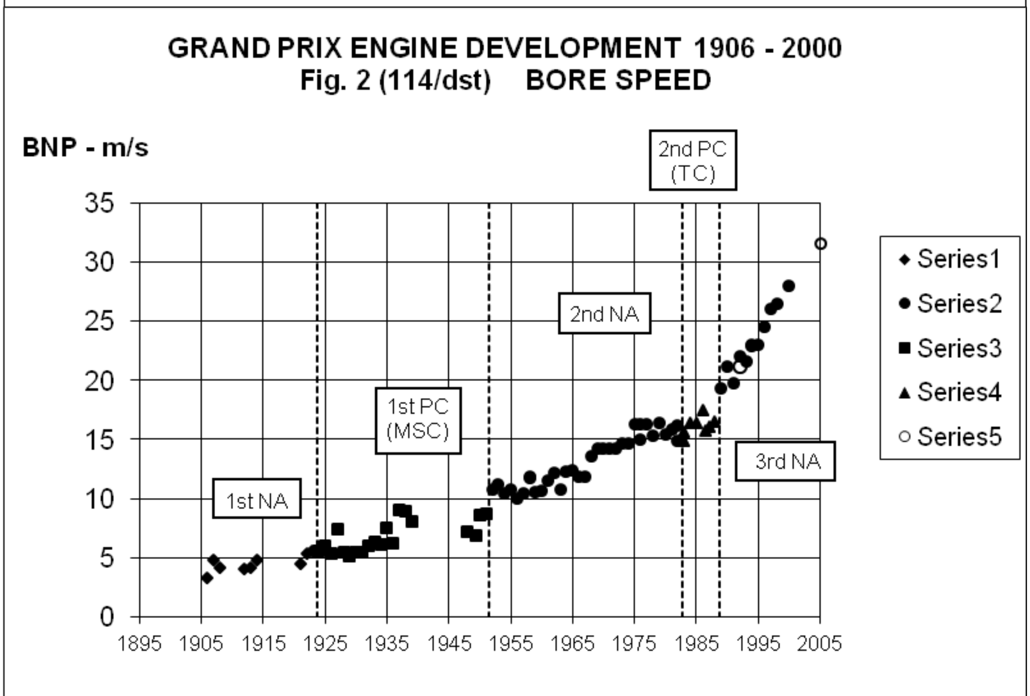
Valve-Spring problems and their solution

Note 13 Part III describes the general theory underlying valve gear, with the parameter Mean Valve Speed (MVS: MVSP at Power Peak RPM NP) of considerable importance. As a surrogate, since the Inlet Valve Lift (IVL) and Inlet Opening Duration (IOD) elements of MVSP are often unavailable, Bore Speed (BNP) can be used since this is dependent approximately on the same material and gear design parameters.

Until 1990 it can be taken that nearly all valve gears had cam-opening with a steel wire Coil Valve-Spring Return System (CVRS), with a few exceptions which will be identified below.

Figs. 1 and 2 illustrate the variation of MVSP and BNP, respectively, over the review period for CoY engines (the definitions of MVSP and BNP are given below).





MVSP = (IVLmm x NP RPM)/(83.333 x IOD^0) m/s; BNP = (Bmm x NP RPM)/60,000 m/s.

Pre-WW1: similarity of BNP

Sub-Note A extends the Fig. 2 data for BNP to a variety of other pre-WW1 racing engines of very different types of Overhead Valve (OHV) gear. It seems very clear from the table that it was BNP which was controlling engine speed at that time, varying from a 10 sample average of 4.1 m/s by +17% to -13%, a range from lowest to highest of x1.34. By contrast, Mean Piston Speed (MPSP) is shown to range from 7.3 m/s to 17.0, i.e. x 2.33

The similarity of BNP over 9 years of pre-WW1 OHV engine development of such disparate types is due presumably to problems caused by the crude cam shapes and the low fatigue life of the spring materials of the time. The Henri-designed Peugeot Double Overhead Camshaft (DOHC) designs, although *fundamentally* the "Way-to-Go" – by minimising valve gear inertia – nevertheless did not then represent an advance in MVSP. Specifically, his 1914 engine did not reach the same BNP as the rival Mercedes Single Overhead Camshaft (SOHC).

Where known for this period, MVSP was around 1.3 m/s.

Overhead Camshafts post 1912

Post the 1912 Peugeot introduction of DOHC all CoY engines were either built to that system or SOHC, with the single exception of the 1936 Auto Union (Eg. 22) which had Push-Rod Operated OHV (PROHV) for its exhaust valves (and a consequent inability to over-rev for a passing opportunity (607)); SOHC nearly disappeared in 1931 when Bugatti (via Miller) finally saw the light with the T51 (Eg. 17), except for the Australian Repco engines of 1966-1967 (Egs. 45 and 46).

Inter-War MVSP improvements

Post-WW1 there was a rapid increase in MVSP to around double the pre-WW1 level. Increased understanding of cam profile effects and of spring design produced this improvement. For example, the 1921 Duesenberg IL8 3 L (Eg. 7) used to break springs at MVSP = 2.1 m/s until Elbert Hall (involved during the war with the "Liberty" aero-engine design) redesigned the cams (6). Not only were the failures overcome but there was also a 16% power increase from the higher RPM tolerable.

The early '20s FIATs (Eg.8 1922 IL6 2 L) also, although very successful, suffered many spring failures (25). They had triple concentric springs to guard against this causing a valve to drop into a cylinder (4). When Vittorio Jano moved from FIAT to Alfa Romeo to design the 1924 P2 IL8 2 L (Eg. 10) he found that the way to prevent the spring surge which was causing the failures was to wind the coil springs with fewer turns of thicker wire ("surge" being the undamped resonant vibration of the middle moving mass of the coils). This raised the spring natural frequency.

Outside Grand Prix racing, Rolls-Royce found that when they uprated the RPM of their 1929 R-type Schneider Trophy-winning engine by 10% in 1931, to about MVSP = 2.6 m/s, they also ran into surge failures and they solved them as Jano had done. They described the resultant spring design as "somewhat reactionary" (1015)!

This MVSP around 2.6 m/s was about the level reached reliably by GP engines before WW2. The 1938 Mercedes M154 60^0V12 3 L (Eg. 24) reached 3.0 only after a great deal of experimentation with the gear and it was reduced to 2.5 m/s in the 1939 M163 (Eg. 25) when the engine was run more slowly (possible with 2-stage Roots supercharging) and IOD was increased (468).

Pre-WW2 Hairpin Valve-Spring Return System (HVRS)

The Hairpin Valve-Spring Return System (HVRS) was pioneered by a Sunbeam motorcycle in 1925, and by the '30s most British racing motor-cycle engines had changed over from CVRS to HVRS. In this system the spring material is in tension instead of torsion to obtain freedom from surge and it also provides isolation of the spring from the hot valve stem (the hairpins were often exposed to the cooling airstream as well). The springs were much more expensive, having to be hand-made instead of machine-wound and they were also much bulkier. The 1938 Norton 1 cylinder 500cc aircooled TT-winning engine with DOHC (SO12) had MVSP = 3.1 m/s and BNP= 9.2 m/s

Post-War HVRS

When Gioachino Colombo designed 60^0V12 engines for Enzo Ferrari immediately after WW2 he accepted the advantages of HVRS over CVRS, as the latter then existed. However, the level of MVSP reached, even by his late-1949 DOHC-redesigned 60^0V12 1.5 L (Eg. 27) was only 2.3 m/s. The engine RPM were restricted by breathing so the extra expense of HVRS was unjustified. When Aurelio Lampredi also used it for the Naturally-Aspirated Ferrari type 500 IL4 2 L of 1952 – 1953 (Egs. 30, 31),

MVSP was pushed up to 3 m/s, i.e. the same as the 1938 Norton of identical individual cylinder capacity. These Ferraris were the last-but-one CoY to use HVRS. Two Grand Prix engines designed originally in 1953 with HVRS, the 90^0V8 2.5 L units of the Lancia D50 and the Coventry Climax FPE were both re-designed to CVRS.

Desmodromic Valve Return System (DVRS)

In 1953 Mercedes-Benz adopted "Desmodromic" , mechanically-closed, valves for their new M196 IL8 2.5 L engine (Eg. 32), the 1st DVRS since Delage tried it without success in 1914 (see Sub-Note B). It was mainly to secure protection against valve-to-piston collisions if/when oversped in the heat of battle but also to obtain sufficient opening area to produce over 100HP/L while retaining IOD of only 256^0 to assist flexibility. This DVRS reached over 5 m/s (468). The problem of differential thermal expansion, which had defeated all previous DVRS designs, was solved serendipititiously by leaving a small clearance between valve and seat to be closed by charge compression pressure. Cost was high, which was not a problem for Mercedes but probably was a reason why only one subsequent GP engine ever raced with DVRS, the American IL4 2.5 L Scarab copied from the M196, and it was unsuccessful (see Sub-Note B). Many other companies did experiment with the system, including Cosworth in the mid '70s. The Ducati motor-cycle racing engines were fitted with DVRS successfully by Fabio Taglioni in 1958 and that firm have retained it ever since for both racing and production engines but they are also unique in their sphere.

Improved CVRS

The coil valve spring received a boost in fatigue life in the mid 1950s from the availability of Swedish wire of very high material purity, i.e. less slag inclusion. The process of shot-peening was also added to provide a compressive anti-fatigue surface layer. This wire spelt the end of HVRS where it had been adopted, eg in Ferraris from 1946 to 1955. The last CoY HVRS was the 1958 Constructors' Champion Vanwall IL4 2.5 L, which was a carry-over from its Norton "Top End" origin. Having also used HVRS from 1947 in their 135^0V16 1.5 L, BRM – unfortunately very far from CoY – finally converted their IL4 2.5 L to CVRS in mid-1960.

Torsion-bar springs

An exception to both systems, also non-CoY, was the 1966 Honda 90^0V12 3 L which used torsion-bar springs, i.e. directly not coiled. Although unsuccessful in GPs, the Honda IL4 1.0 L Formula 2 engine with the same feature was nearly unbeatable in 1966.

Double Coil Springs with interference fit

Another step forward for CVRS in the mid-'60s was the adoption of selectively-fitted interference between outer and inner springs – dual springs having long been used but with no touching of the coils. This provided damping to reduce surge amplitude. Rolls-Royce used this idea in their 1964 IL6 4.0 L FB60 engine for the BMC "Princess" (865). The Ford "Four Cam" Indy 90^0V8 4.2 L engine (SO17) was so assembled in 1965 (864) with MVSP = 3.8 m/s and the Cosworth DFV 90^0V8 3 L DFV (Eg. 47) was a notable user of the principle some years later. Because of the inevitable wear reducing the damping with running the method was restricted in high-speed engines to short life and was therefore not suitable, eg, for Le Mans 24 Hours (16) (the pioneering FB60 was of course not a high-speed engine and also the inner spring was flat section to reduce wear). In Grand Prix racing the springs had to be replaced after every event (59).

Cosworth DFV development

Although held back for 3 years by a valve-drive resonance, at the end of its development in 1982 the Cosworth DFV had reached MVSP = 4.3 m/s typically but up to 5 m/s when care was taken to avoid overspeeding (Eg. 62).

Pneumatic Valve Return System (PVRS)

In 1984 Jean-Pierre Boudy and his employer Renault applied for a French patent on a valve return system using gas as the springing medium (474). The conventional valve coil spring was replaced by a piston on the stem in a fixed cylinder containing compressed gas which was compressed further as the valve was opened by a cam in the usual way. Thereby energy was stored in the gas to return the valve to its seat. Renault named this system "Distribution Pneumatique" (DP) but when it was taken up by Honda later they used the description "Pneumatic Valve Return System" (PVRS) which is more precise. Renault put this DP into their 1986 TurboCharged EF15bis 90^0V6 1.5 L and claimed a 14% increase in useable RPM (596).

General adoption of PVRS and its advantages

There was no immediate CoY success for the new Renault EF15bis engine. The company abstained from Grand Prix racing during 1987-1988 but retained DP = PVRS for their entry into the new 3.5 L Naturally-Aspirated formula in 1989. Championship success was slow in coming but, by 1991, it was clear that they had once again pioneered the next major step in racing engine design and it was then generally copied.

The particular advantage of PVRS was given in (468) where Ilmor reported that, in moving to it for their type 2175B 72^{0}V10 3 L in August 1994, the pressurised gas column had a natural frequency x8 that of a comparable wire coil spring, so putting surge far beyond the operating RPM. As a serendipitous benefit weight – Top-End weight - was reduced. When Brian Hart in 1997 adopted a proprietary Del West PVRS for his 1996-design type 830 V8 3 L, he saved 5kg, about 5% (567).

The need for costly coil valve spring replacement after every event was also abolished.

The gas used in PVRS was usually nitrogen, topped as necessary from an on-board bottle, but the 1999 Ilmor engine had an air compressor to supply the system.

Titanium-alloy Valves: US V8s and Japanese motor-cycles

Far outside the Grand Prix sphere and well ahead of it in time, in 1962 American engine tuners of their large V8 production units with Inlet Valve head Diameter(IVD) up to 2¼ inches, opened by pushrods (PROHV), had reduced their valve gear problems by adopting for the inlets the alloy Ti6Al4V (titanium alloyed with 6% aluminium and 4% vanadium) of only 60% density compared with steel (220). This material was originally a USAF-financed development in the 1950s for jet aero engines and the Californian firm Del West pioneered it for auto racing engines. This material change was often teamed with rifle-drilled hollow valve stems to save more mass.

In 1983 Honda for the 1st time used Ti-alloy valves both for inlets *and exhausts* in their 90^{0}V4 850cc sports-racing motor-cycles (97). This was satisfactory for a short life. Figures given later for the Yamaha YZF-R7 IL4 750cc for the same racing category showed the practical mass reduction as follows (656):-

	Inlet	Exhaust
(Ti-alloy valves / Normal valves)	0.52	0.56

(The figures imply that the Ti-alloy valve stems were also drilled

Application of Ti-alloy valves to Grand Prix engines

During the TurboCharged era, 1983-1988, it was not possible to cope with exhaust valve temperatures using Ti-alloy and, without having *all* the valves in the lighter material, there could be no gain in MVSP because the heavier exhaust valves would become the valve-bounce-limiter. When the Grand Prix formula reverted to Naturally-Aspirated in 1989 engine designers then adopted all-Ti-alloy valves. The figures for the (non-CoY) Cosworth HB series, as developed, compared to the preceding steel-valve DFR type illustrate the advantages gained (they have to be shown in BNP because internal details are not available):-

	1988	1989	1991
Engine type	DFR	HB2	HB5 (pre-PVRS)(743)
BNP m/s	16	18	20
	Datum	+12%	+25%

Combined PVRS, All-Ti-alloy-valves and DLC

By 1992 the “Dream Design” of “PVRS + All-Ti-alloy-Valves” was well established. Figures 1 and 2 show that, from then onwards, there was a more rapid rise of MVSP versus time than ever before as designers learned to exploit this combination (limited data but over 10 m/s in 2000) and of BNP (28 m/s in 2000) (details from the Ferrari type 049 90^{0}V10 3 L). In particular the new “Diamond-Like-Carbon” (DLC) very hard, low-friction surface treatment was essential to withstand the contact stresses and rubbing speeds of the valve gear (Note 103).

Post-2000 data

Post this review to the end of 2000 some non-CoY data has been released which seems to show that valve-gear development has reached a peak, as follows:-

Date	Engine	Data Source*	MVSP m/s	BNP m/s
2005	BMW P85**	(1095)	na.	31.5
2006	Cosworth CA/6	(1069) (1092)	na.	31.4

After the regulations imposed Maximum RPM, the BNP figure fell back:-

2009	Toyota RVX-09H Maximum RPM limited to 18,000.	(1091)	10.0***	28.0

*Some sources additional to Appendix 3:-

(1069) Race Engine Technology No. 20 Jan/Feb 2007 Article by S. Corbyn.

(1091) Race Engine Technology No. 49 Sept/Oct 2010 Article by I. Bamsey.

(1092) Cosworth website 6 October 2010 for CA2010.

(1095) *Ten Years of BMW F1 Engines* M Theissen et al 2010.

**An advanced engine tested but not raced due to a change of life rules. Plotted on Fig. 2.

***IVL given but IOD = 320^0 assumed.

Possible limit to PVRS

It may be that the piston/cylinder sealing materials of PVRS have limited the BNP achievable. Ref. (988) stated that in the 2003 BMW P83 the temperature in the gas reached 250C.

Note 15 Sub-Note A

Wider sample of Bore Speed for pre-WW1 OHV engines

Date	Engine	Data Source	BNP m/s	Type of OHV PR = Push-Rod S = Single / D = Double: Over-head Camshaft OHC VIA = Valve Included Angle	MPSP m/s
1906	Sizaire	(259)	4.00	PR Inlet + Suction Inlet	7.33
1907	Sizaire	(259)	4.00	Presumed OHV	12.00
1907	FIAT GP	Eg. 2	4.80*	2 Valves/cylinder (2v/c) PR push/pull. VIA = 90^0	8.54*
1908	Sizaire	SO3	4.00	Presumed OHV	20.00
1908	Mercedes GP	Eg. 3	4.13	PROHV Inlet (Side exhaust)	9.60
1910	Lion Peugeot	SO4	2.93**	SOHC Inlet(Side exhaust)	20.53
1910	FIAT S61	(4)	3.58	4 v/c SOHC. VIA = 0	10.45
1911	FIAT S74	(519)	4.00	“	10.67
1912	Peugeot L76	Eg. 4	4.03	4 v/c DOHC. VIA = 60^0	14.67
1913	Peugeot L54	Eg. 5	4.17	“	15.00
1914	Peugeot L45	(485)	4.29	“	15.68
1914	Mercedes M93654	Eg. 6	4.81	4 v/c SOHC. VIA = 60^0	17.05

Average of 10 engines, excluding 1907 FIAT GP* and 1910 Lion Peugeot**

4.1 (+17%; -13%)

Highest/Lowest BNP = 4.81/3.58 = 1.34 MPSP 17.05/7.33 = 2.33

* and ** See next page.

*Suspected to be too high on quoted NP because (4) gives independent ACF test evidence that the original 1905 basic design had BNP = 3.3 m/s. It is unlikely that the cumbersome OHV could have been improved by 45% in 2 years, especially as the power gain was quoted at only 8%. A gain of this amount from RPM would mean BNP = 1.08 x 3.3 = 3.56 m/s.
**Note that Inlet Valve head Diameter/Bore (IVD/B) was 0.77!

Note 15. Sub-Note B
Other DVRS in Grands Prix

1. 1914 French GP: Delage

Designed by Arthur Michelat, the details of the 1914 type S 4 v/c DVRS were concealed by misinformation fed to the press at the time and only revealed in 1986, when a survivor in the USA was dismantled. It included small final-seating springs (597A). It was IL4 4.5 L 94mm/160 = 0.59 (485). The engines were down-for-power in the race after a last minute modification to their valve gear according to (940), which prevented proper seating according to (1047). Only 1 car finished out of 3 in 8th place. Clearly, DVRS had not been mastered and Delage did not use it for their successful post-WW1 2 L and 1.5 L engines.

2. 1960 Scarab

Leo Goossen, *the* American racing-engine designer with a career over 53 years, prepared in 1959 an IL4 2.5 L 3¾ inch/3⅜inch = 1.11 unit for Lance Reventlow's Woolworth-fortune-financed Scarab team. He was told to fit 2 v/c DVRS (1046) copied from a 1955 Mercedes-Benz 300SLR (M196I) given to the Ford museum at Detroit, although the clearance adjustment method differed from the M196. After some initial valve gear trouble (1046) the car, with a front engine inclined at 11^0 to the horizontal (another feature similar to the M196), was entered in five 1960 GPs but qualified only twice because of engine failures or being just too slow. It obtained one 10th place. The 2.5 L formula and the money then both ran out.

The IL4 Scarab and donor IL8 Mercedes DVRS compared as follows (both 2.5 L):-

	1955 M196 GP	1960 Scarab
Data Source	Eg. 33	(943,1046)
MVSP m/s	5.2	4.5
BNP m/s	10.8	11.9

Illustrations

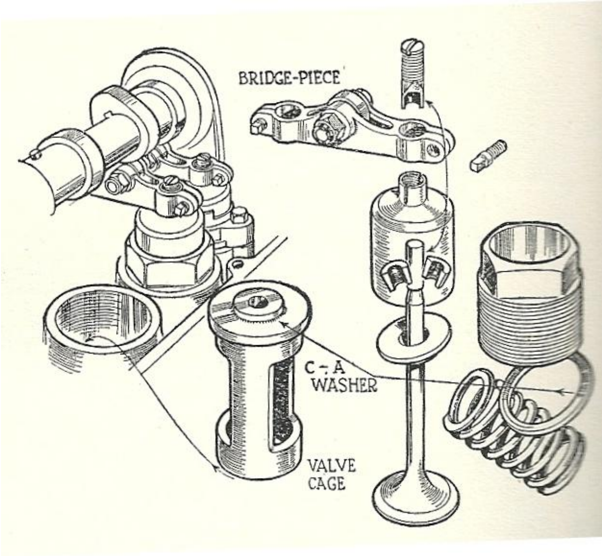
An Appendix gives illustrations of the various types of valve gear.

Appendix

Coil Valve-Spring Return System (CVRS)

There are, of course, many examples of CVRS but an early version applied to OHV for which an illustration is available is the 1911 10.1L FIAT S61shown here.

DASO 4

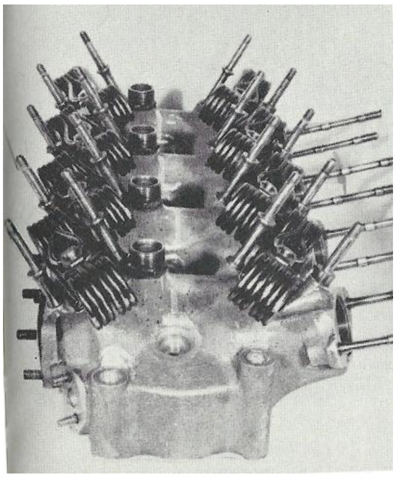


This engine also had possibly the 1st inverted-cup tappets to resist cam side-thrust (see Note 25B).

Hairpin Valve-Spring Return System (HVRS)

The last CoY engine of only 4 to use HVRS was the 1958 Vanwall V254 shown here (the other 3 were:- the 1949 DOHC 1.5L Ferrari; the 1952 & 1953 2L Ferraris).

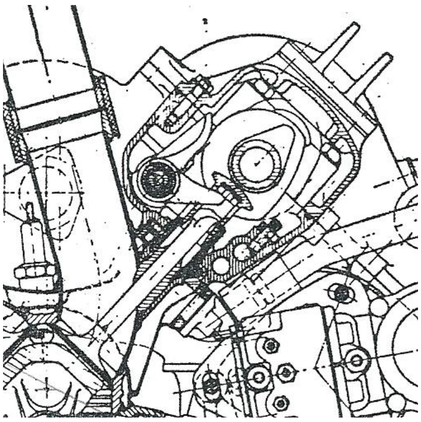
DASO 68



Desmodromic Valve return System (DVRS)

The only CoY engines to use DVRS were the 1954 & 1955 2.5L Mercedes-Benz M196s.

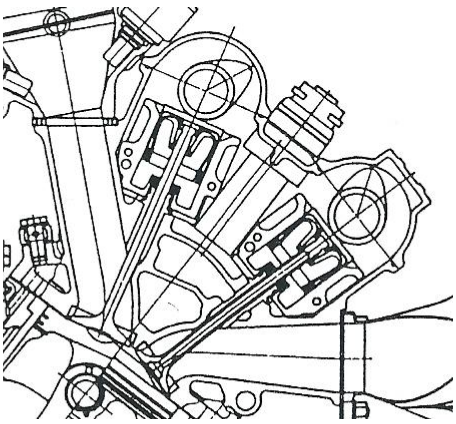
DASO 468



Pneumatic Valve Return System (PVRS)

The illustration is of the 1992 3.5L Honda RA122E/B.

DASO 69



Note 16

Sparking-plug Problems and their solution

Up until 1939 sparking-plugs were often blamed unjustly by team managers for racing retirements but actually the basic problem of keeping the high-tension central electrode insulated while getting rid of the heat absorbed by the surrounding insulation had been solved before WW1 by using mica for the latter instead of the brittle porcelain of the original de Dion-Bouton type used from the start of the 20th C (520). The 1914 Grand Prix Mercedes used mica-insulated plugs after the original type selected (which may have been porcelain) proved unsatisfactory at high RPM (468). The fact that 3 plugs were fitted per cylinder in that engine indicated partly, no doubt, Daimler's "Belt and- Braces" philosophy – they actually provided bosses for 4 per cylinder! – but also the state of plug design at that date. By coincidence this M93654 Mercedes is the 1st Grand Prix unit for which a contemporary power curve is known but it may not have been a coincidence with the plug change that it was tested over the natural power peak (468). This was not usually the case with contemporary non-GP automobile and aero engines.

The feature of 3 plugs per cylinder recurred in Harry Ricardo's 1922 3 L TT design for Vauxhall (242) but only 1 was used (4).This may mark the date at which confidence in plugs was well established. Aero piston engines, of course, used 2 plugs per cylinder from an early date and continue to do so to this day, as a matter of safety.

Temperature variation problem

A particular plug problem remained for racing engines. A plug with a large diameter heat-path to coolant from the nose of the insulator which, at full power, ran comfortably below the temperature (about 850C (942)) which could cause pre-ignition of the mixture would not, at warm-up speeds, burn off the short-circuiting deposits from oil reaching the combustion chamber through piston clearances above the operating figure (294). The range of heat dispersion to be covered was, of course, magnified in proportion to any pressure-charging.

This problem was avoided by warming-up on "hot" (or "soft") plugs with smaller diameter heat paths (724) and then changing to "cold" (or "hard") plugs for the race. On occasion overheated plugs in an over-revved engine could still cause a pit-stop for a change or a slow-down to let them cool, see Sub-Note A.

Optimum plug temperature was 550C (942).

Sintered aluminium oxide insulation introduction

Just before WW2, to meet the needs of supercharged military aero engines running on highly leaded fuel, where the combustion products attacked mica, a stronger ceramic insulator was developed. This was sintered aluminium oxide, trade- named "Corundite" by KLG, the original makers. This material had 2x the strength and 5x the thermal conductivity of the best porcelain and the new plugs using this insulation had 4x the heat range of mica plugs (76,611). It became standard post-WW2 for motor-racing. The warm-up/plug-change routine was then not always necessary (52) but was still used sometimes, e.g. by Mercedes-Benz (conservative again) in their 1954-1955 M196 (613). [In 1951 the Bosch 14mm plug range had a "softest" plug (W240) with a 5.5mm diameter at the insulator nose; the "hardest" plug (W440) had a 9.5mm nose (ref. *Autocourse* 1951/2, article by Bosch)].

By 1968 the individual plug heat range had certainly reached a level at which only one type needed to be used in a naturally-aspirated (NA) GP engine (634).

Two plugs per cylinder

Two plugs per cylinder appeared in the 2nd naturally-aspirated (2NA) era, post 1951, as Bore/Stroke ratio increased with 2 valves per cylinder where the ideal central location for one plug was not available. This duplication was to aid combustion rather than reliability.

Four valves per cylinder and reversion to single plugs

The return of 4 valves per cylinder in 1966 freed the efficient central location and a single plug has been used there ever since.

Plug size

The size of plug has come down in 3 steps, hence reducing the heat absorbed which has to be dispersed, reducing the restriction on overhead valve diameter and improving its placement towards the cylinder centre in 2 valve/cylinder engines and, finally, reducing its weight.

Originally an 18mm outside diameter (OD) of screw thread size was chosen by de Dion-Bouton and this became the European standard (428). A reduction to 14mm OD was possible in the mid-30s (294) and became general in post-WW2 designs. A 10mm OD plug appeared in the early '50s and is still used as standard. Honda, with their miniaturised 5-cylinder 125cc racing motor-cycle in 1965-1966 actually fitted an 8mm OD plug in a 35.5mm bore with 4 valves per cylinder (76).

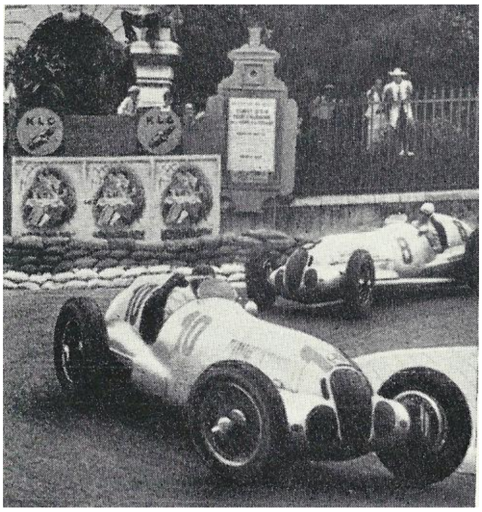
Note 16. Sub-Note A

Sparking plug overheating in supercharged racing engines

During the famous 1937 Monaco Grand Prix duel between Mercedes-Benz W125 team-mates Carracciola and von Brauchitsch – the latter completely ignoring the manager's pit signals to stop hounding or let pass his leader! – Carracciola certainly had to have his plugs changed and (280) times it as *after* raising the 1935 circuit lap record by 11% at 74% distance. A more detailed history (887) puts the plug change much earlier when he had lapped at "only" 9½% faster than the record.

The picture shows the two "team-mates" during their Monaco scrap,
von Brauchitsch leading Caracciola.

At any rate the spectators were saved from a boring procession, even if Neubauer was having apoplexy!



Another example

During the 1939 Eifelrennen Lang in a Mercedes-Benz W154/M163 fitted with the 1st 2-stage supercharger (468), in order to pass Nuvolari's Auto Union, over-revved in 4th gear above the 7,500 RPM red-line to 9,200 (according to Neubauer (886)). This overheated his plugs which caused misfiring (pre-ignition from incandescent insulators) but by easing off to cool them he was able to continue and win the race (612).

Note 17
Exhaust Valve Problems and their solutions

Early Engines

When Maybach in 1885 designed a 264cc water-cooled 1-cylinder engine for Daimler, he placed the exhaust valve at the side of the cylinder as the obvious way to operate it directly from a camshaft alongside the crankcase, itself driven easily from the adjacent crankshaft. It was probably about 24mm head diameter (EVD). The maximum stress in a 4-stroke poppet exhaust valve – in the stem – need not be high (depending on the $(Head/Stem)^2$ ratio) and it is compressive while under cam control at opening and closing, being tensile while the spring is controlling the valve in between. The stress range is repeated at RPM/2 so that material fatigue life at high temperature – up to 800C without internal cooling – is important, as is creep life and resistance to attack by combustion products. The material available in 1885 was only steel and probably a mild carbon type was used (say 0.2% C) as would have been used in boiler-tubes exposed to firebox gases. The side-valve had the advantage that a lost valve head would not necessarily wreck the engine (and this was a favourable design point considered in production engines and some racing units for a further 4 decades).

Nickel Steels

Shortly after the early Daimler engines appeared, Riley in 1889 (618) published the work on Ni/Fe/C alloys which opened the way to greater life, or greater stress for the same life, at high temperatures. Advantage was taken of this, inter alia, for the manufacture of gun barrels (a gun being a piston engine which ejects its piston after the power stroke!). Krupp of Germany in particular exploited this use (and also employed it for battleship's armour plate). It is not known specifically what use was made of these Ni/Fe/C alloys for exhaust valves at the time, but in 1916 Ricardo specified a 3% Ni type for his 1st tank engine – one of the few parts for which he was not restricted by low supply priority to low-grade steels (242).

Wright Aero-Engines

When the ingenious Wright brothers built their own 3.3L 4-cylinder aero-engine in 1903, not having found a proprietary unit of sufficient Power/Weight ratio, they used originally cast-iron heads with steel stems screwed–in (their exhaust valves moved vertically below a horizontal combustion chamber which was only slipstream-cooled). Many valve failures were experienced (the location here also must have alleviated the resultant damage) and, despite several engine developments, continued over the next 10 years (592,616).

Stainless Steel

In 1912/1913, while experimenting with Cr/Fe/C alloys for gun barrels, Brearley (head of the Brown-Firth collaborative laboratory) discovered accidentally the stainless (i.e. oxidation-resisting) properties of a 12.8%Cr/86.96%Fe/0.24%C mix (hereafter figures only; refs 618,629,669). This then offered superior resistance to scaling at exhaust temperatures.

Valve Size Advantage

Meanwhile in engine high-power design the breathing and valve-bounce-speed advantages of 4 overhead valves per cylinder (4v/c) had become clear and this also assisted the life of exhaust valves. This is because, with a fairly-constant thickness of valve-head to minimise mass and so maximise valve-bounce RPM, the smaller the diameter the better the ratio of rim flow area through which much of the heat escapes during the valve-shut period (2/3rds of the time with early engine timing) to the head area by which the heat enters the valve (see Sub-Note A). Whereas a typical 1908 12L 4-cyl. engine with 2 v/c would have had an EVD around 100mm, the 1912 7.6L 4-cyl. Peugeot with 4 v/c was down to 54mm and the 1914 4.5L 4-cyl. Mercedes, also 4 v/c, to 43mm. Although Fiat led a return to 2 v/c in 1921, their "GP Car-of-the-Year" in 1922, being only 2L and 6-cyl., had EVD = 40mm so was not breaking new ground there.

WW1 Advances

It is interesting, in looking at this valve size aspect, that, when Henry Royce designed his 1st aero-engine of 20.3L in 1914 and retained the 4.5 inch (114.3mm) bore he knew so well from the 1907-onwards 40/50 HP ("Ghost") production car, his new overhead-valve design (based on the 1912 DF80 Daimler 7.3L 6-cyl. aero-engine which had run in a car in the 1913 Sarthe GP, an example having been obtained by R-R, (305)) required an EVD of only 54.8mm. This engine (later named "Eagle"), though containing 50% more parts than any other in British WW1 service, was "undoubtedly" the most reliable so that an overhaul life still only about 100 hours was nevertheless nearly x2 that of the next-best (242). Ref (616), referring to WW1 aero engines in general, states that "exhaust valve failures were probably the most prevalent cause of ….short engine life". The exhaust valve material of the Eagle is not known but (669) states that high-Cr steels were used in WW1 for aero-engine exhaust valves – a "Firth Aeroplane Steel" (FAS) was available in 1914 with 13Cr/86.6Fe/0.4C.

Ref. (616) reports , without mentioning applications, that during WW1 alloys up to 25Ni/Fe/C were tried to combat exhaust gas attack but were found to have poor creep resistance.

The late-WW1 US "Liberty" 27L V12 engine, 2 v/c, used "High Speed Steel" (HSS) valve material derived from the early 1900s cutting-tool alloy of 19W/4Cr/Fe/C developed by Taylor and Wright in America (618). While this had the necessary strength at high temperatures, it had poor corrosion resistance (616)(of course, in cutting-tool use the material would have been cooled externally with soapy-water). The Liberty valves were 69.9mm EVD and the head was very thin (399). The engine had a short overhaul life when it eventually entered service.

"Tungsten Steel" – which must mean HSS – actually had been used previously by Pomeroy (Senior) for the 1914 GP Vauxhall exhaust valves (224). It was also used by Halford and Pullinger for the exhaust valves of the 1916 BHP aero-engine, redesigned from the original Porsche-designed pre-WW1 Austro-Daimler unit built under licence by Beardmore. The BHP also took advantage of size effect by changing a single 71mm valve to dual (602).

Ref. (669) also reports that Krupp, just before WW1, had introduced V2A containing 20Cr/7Ni/72.8Fe/0.2C. It is likely that this was used in German engines. The Cr + Ni content made this an "Austenitic" steel, so-called because the grain structure at operating temperatures was similar to a well-known Fe/C metallurgical phase named after Robert-Austen, a late-19th C researcher (618).

<u>Austenitic Steels</u>

Post-WW1 the Versailles Peace Treaty forbade the manufacture of German high-power aero-engines. Perhaps because of this in 1923 Krupp sold rights to their patented austenitic steel to Firth and Brown, possibly on the initiative of Brearley. These firms developed their own versions (669). In particular Hatfield, Brearley's successor at the collaborative laboratory, produced the now-famous 18Cr/8Ni/73.8Fe/0.2C stainless-steel, sold by Firth's as "Staybrite".

The problem of the exhaust valve was brought under control a little later by a revised formulation which balanced Cr and Ni and added some W and other elements. The originator is not given in (669) but an early example was Kayser-Ellison KE965 with 13Cr/13Ni/2.5W/1.3Si/0.7Mn/69.1Fe/0.4C (with the usual important maximum controls on impurities such as S and P and the appropriate heat treatment). This alloy was specified, more or less, by Air Ministry DTD49b and also by En54 (639).

This proprietary alloy KE965 was the exhaust valve material of 1st choice in British high-power designs for the next 3 decades. Craig (no doubt referring to British makes) asserted that austenitic material was used initially in air-cooled racing motor-cycle engines and then was taken up for aero-engines (12).

<u>Internal Cooling</u>

At about the same date that the new austenitic exhaust valve alloy was available, a further step in valve design had been taken at Daimler which became nearly-standard practice when engines were Pressure-Charged (PC). This company had experimented with PC during WW1 to restore power at altitude and had settled on mechanical supercharging (MSC) with the Roots blower (637,468). Post-War they applied MSC as a short-term power-boost system to various 4-cyl. touring and racing cars. When Porsche joined them in 1923 he improved the existing M7294 IL4 2L MSC engine by, amongst other things, fitting exhaust valves whose larger-diameter stems were drilled and then plugged after partial filling with a quantity of Hg. When the valve reciprocated this filling was shaken up-and-down the stem and so conducted heat more rapidly from the solid head to further up the stem from whence it could escape via the guides to the cooling water (468). This revised engine won the 431 km 1924 Targa Florio. Porsche in the meantime had designed a completely-new unit with continually-engaged higher-boost supercharger, the 1924 M218 IL8 2L Grand Prix engine in which the stems of all-4 v/c were filled partially with Na salts (468). This filling liquefied at operating temperature and wetted the metal surfaces, unlike Hg, and so conducted heat away more effectively.

The exacerbation of the exhaust valve problem by PC was solved not only by improved austenitic steel and internal cooling, as described above, but also by the near-simultaneous adoption of alcohol-base fuels, i.e. fuels whose latent heat of evaporation was a multiple of that of petrol, some of which entered the cylinder neat and reduced metal temperatures (see Appendix 2).

In aero-engine research of the mid-20s, using petrol fuel and with many firms building air-cooled types, i.e., with hotter cylinder heads than water-cooled designs, it was Heron (an Englishman, ex Royal Aircraft Factory in WW1) with the US Army Air Corps who followed Midgely in developing internal Na-cooling of exhaust valves (616). These valves were then produced in quantity by the Williams Rich Corporation (708). At a critical moment in the British preparations for the 1931 Schneider Trophy air race, a sample American Na-cooled valve was brought to England by Banks and given to Rolls-Royce to copy under sub-licence from the Bristol Aeroplane Co., who had a licence from the Rich Corpn. (ST competitors had to use nationally-made parts)(619). When fitted into the R-type engine and run with the 20% petrol/70%benzole/10%methanol fuel at the mixture strength necessary for the Supermarine S6B seaplane to perform competitively within the rules, Lovesey, the development engineer, was able to see in the open exhaust ports that the "Bright Cherry Red" of the uncooled 50.8mm parts (about 850C, which would pre-ignite the charge) was changed to "almost black" when the internally-cooled valves were fitted – probably 300C cooler (615,689).

This type of valve, in KE965 alloy (also used for solid inlet valves*) was later included in the Rolls-Royce "Merlin" and "Griffon" aero-engines, produced in vast numbers during WW2 (632).

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*Surplus Merlin inlets were turned-down post-WW2 by motor-cycle racing men to serve as exhaust valves, e.g., in Excelsior "Manxman" 250cc!

Sodium-cooled exhaust valves – or, in the case of the Mercedes-Benz 1934-1939 M25 to M165 engine series, Na-, and then a reversion to Hg-cooling in hollow-stem Krupp WF100 valves (15Cr/14Ni/2W/2Si/0.8Mn/65.75Fe/0.45C, very like KE965)(30) – were virtually standard for supercharging to the end of the 1st GP PC era in 1951. However, it is noteworthy that MG in 1936 were able to run *solid* KE965 valves (810) of only 28.6mm EVD at an absolute pressure at the inlet valve (IVP) up to 3 Atmospheres (ATA) in record tune on high-alcohol mixture, once the valve guides had been lengthened in an ad hoc way to improve heat escape (139,394). This shows the inverse size advantage when combined with the parts-cooling effect of alcohol.

Fuel Cooling

During the latter part of the 1st PC era – all engines mechanically supercharged – Grand Prix HP/Litre was raised by taking advantage of this ability with alcohol fuel to run super-rich (i.e., compared with the stoichiometric ratio) so that, as well as cooling the charge in the inlet manifold by evaporation, some fuel passed as liquid into the cylinder and there cooled by final evaporation the valves and piston. The cost of this on consumption has been estimated as follows:-

Data Sources:- 31,607,684

Year	1937	1938	1951
Car	Mercedes W125	Mercedes W154	Alfa 159M
HP/L	97	158	270
Relative Specific Power	Datum	1.63	2.78
Fuel, %age Alcohol	65 [10 Ethanol] [55 Methanol	86 Methanol	99 Methanol
Relative Consumption Per HP.Hour	Datum	1.65	2.44

Internal Cooling in NA engines

Sodium stem-cooling was used for the exhaust valves of some British air-cooled racing motor-cycles in the last few seasons before WW2 (675). The temperature drop here would not have been necessary to prolong life but would have enabled compression-ratio to be raised. Post-War the same principle was extended sometimes to inlet valves, egs, air-cooled Nortons, Mercedes M196, BRM P56 and Porsche 908 & 912 air-cooled sports-racing engines, where the cooling would also improve breathing. In the M196 of 1954-1955, the directly-injected moderate-alcohol fuel was sprayed partially onto the exhaust-valve head and so cooled it additionally.

In the air-cooled Nortons of 1953-1954 exhaust heat was taken from the valve guide by an oil gallery and then dispersed in a separate oil/air cooler.

WW2 Advances

During WW2 the need for high-temperature creep-resisting materials for jet engine turbine blades led in the UK to the development by the Wiggin company of the Nimonic alloys, which started from the pre-War Nichrome formula (80Ni/20Cr) made for electric-fire elements, i.e., a material which resisted oxidation at bright cherry red temperature but, of course with no stress. By adding small amounts of Ti, Al, Fe and C and heat-treating the creep resistance at useable stress was increased (618). From the early-50s this 1941 turbine-grade Nimonic 80A (75Ni/20Cr/2.4Ti/1.2Al/0.5Fe/0.04C (685)) was used for NA solid exhaust valves. Coventry Climax (amongst others) had them in their FPF engines of 1957-1961, which powered the World Champion Cooper cars in 1959-1960 (34). However, Hassan (Climax Chief Engineer) reported that quite bad seat scuffing and ridging occurred so that tune was lost.

Post-WW2 Advances

Because of the drawbacks mentioned to Nimonic 80A, in the 1961-1966 series of Climax FWMV V8 1.5L engines Hassan adopted for their solid exhaust valves a new austenitic alloy 21-4NS (21Cr/4Ni/9Mn/0.25Si/0.4N/64.85Fe/0.5C, (644)). This work-hardened because of the N content and gave “almost complete” absence of valve failure (34). Its Ultimate Tensile Strength at 800C was 20 tons/sq.in. (309 MPa) compared to 16 (247 MPa) for KE965 and its resistance to scaling was x4 (885). It was much cheaper than N80A.This 21-4NS alloy with an addition of 0.2% Columbium was also used for the famous Ford Cosworth FVA and DFV engines over 1966-1983 (583).

Turbo-Charged Engines

Reversion to PC by Turbo-Charging (TC) over 1977-1988, using inter-coolers to prevent knocking on the regulation petrol fuel, but without the advantage of alcohol fuel to cool the cylinder parts as described above in the 1st PC era, needed a return to Na-cooled valves. The head itself, as well as the stem, probably would have been hollow. The frequent highly-spectacular blow-ups of such engines, always ascribed by commentators to the turbocharger, while no doubt often correct, would in many cases be provoked by a broken valve head entering the turbine.

Titanium Exhaust Valves

In 1989 the regulations once again banned PC. Power increases then demanded higher RPM, calling for higher Bore/Stroke ratio, only possible if higher valve speeds could be accepted. To the latter end lower-density Ti-alloy valves were adopted not only for inlet valves but also, astonishingly, for exhausts. Honda had found it possible to do this in 1983 in their V4 750cc 4 v/c racing motor-cycles, where EVD was only 25mm (97). The alloy used was Ti6Al4V (317),i.e., 6Al/4V/90Ti, a “work-horse” specification commercially-available after having been developed at great expense for the US Air Force for medium-temperature gas turbine applications in the ‘50s. In motorsport it had been first used to reduce the mass of inlet valves around 2 inches diameter by American tuners of their large V8 push-rod engines in the ‘60s. These valves were made by the US specialist company Del West. The density ratio (Ti6Al4V)/(KE965) = (4.4g/cc)/(8.1g/cc) = 0.54, so that the maximum advantage on Mean Valve Speed would be $1/\sqrt{(0.54)} = 1.36$ (reduced in practice by non-Ti parts in the valve-gear).

The advantage of Ti-alloy for *all* valves had not reached Grand Prix engines before TC, with its higher heat-flow, made it impossible*, but post-TC nearly all makers adopted it. Exhaust valves in the V10 3.5L, then 3L, units which mostly dominated the post-1988 scene began at about 30mm diameter, but they have risen steadily as B/S ratio has been increased further via the adoption of Pneumatic Valve Return Systems (“Air Valves”), being around 33mm in 1998. Del West were still the major supplier (317).

The thermal conductivity of the Ti-alloy is only about half that of austenitic steel (623), which makes its successful use for exhausts somewhat surprising but, of course, their life now has only to be about 2 1/2 hours to cover race-day “warm-up” plus the race before they are discarded at overhaul.

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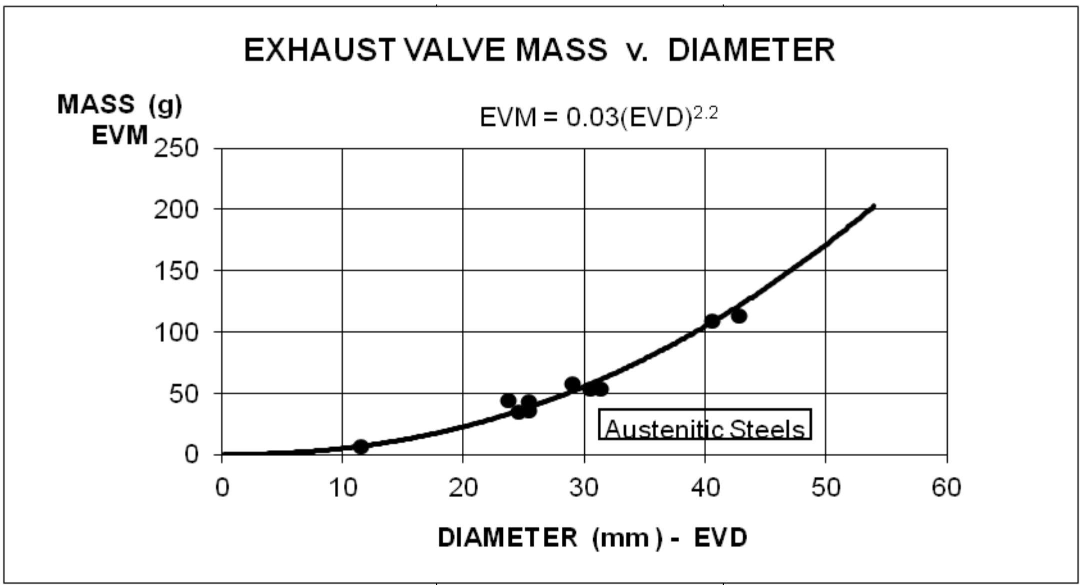
*Porsche actually developed for the TC version of the Type 912 air-cooled sports-racing engine a Ti-alloy Na-cooled *inlet valve (*302). This engine powered the winner of the 1972 CanAm Championship. Apart from retaining the cooling advantages of the previous Na-cooled steel valve, mentioned previously, this might have permitted a more-rapid-opening cam, as well as giving an engine weight reduction.

Sub-Note A
Solid Exhaust Valve
Data

ENGINE	DASO	EVD mm	EVM g
COSWORTH FVA	583	29	58
CLIMAX FWA2	132	30.5	53.1
CLIMAX FPF 1.5	131B	40.6	108.9
CLIMAX FPF 2.5	131B	42.8	112.5
CLIMAX FWM	33	*24.6*	34.6
CLIMAX FWMA	33	*25.4*	35.4
CLIMAX FWMV1	34	*31.4*	53.2
CLIMAX FWMV6	34	*23.7*	44.3
CLIMAX FWMW	34	*25.4*	42.5
HONDA RC149	14	11.5	6

Italics = Approx.



Since the chart shows that EVM is proportional to (EVD)^2.2 and not (EVD)^3, it follows that valve dimensions normal to the head were not *pro rata* to the diameter in the design convention used over this range of 1954-1966 engines. Therefore the (Rim-section Area)/(Head Area) ratio declined with increasing diameter and, other things being equal, the larger the valve the hotter it would run.

One of the (many) advantages of the 4 v/c Ford Cosworth FVA over the 2 v/c Climax FPF, both 4 cyl. of about the same capacity (1.6L v. 1.5L), was that EVD = 29mm compared with 40.6mm (and later 42.8mm).

In "Motor Sport" Jan. 1970 (853), referring to the Ford Cosworth BDA IL4 1.6L (developed from the FVA for Ford road-going high-performance models) it says:-

"*…the smaller valves show less tendency to burn out than the large single valve fitted….*
..to a normal Lotus-Ford…"

The Lotus-Ford was also DOHC IL4 1.6L and the EVD difference was:-

L-F = 36.8mm; BDA = 25.4mm; i.e. 31% smaller.

Note 18

Bearings Development

An attached Table (P.8) details the types of bearings used for the crankshaft Main Journals (MJ) and Crankpins (CP) of successful Grand Prix (GP) engines in the 1906 – 2000 period of this review. They divide broadly into 3 groups, although each group has certain exceptions:-

1st group: 1906 – 1921 Plain Whitemetal;
2nd group: 1922 – 1939 Roller and Ball;
3rd group: 1947 – 2000 Plain "thinwall" copper-lead-indium.

A detailed discussion is given below for each group, after a general introduction.

General Introduction

Plain Bearing Theory

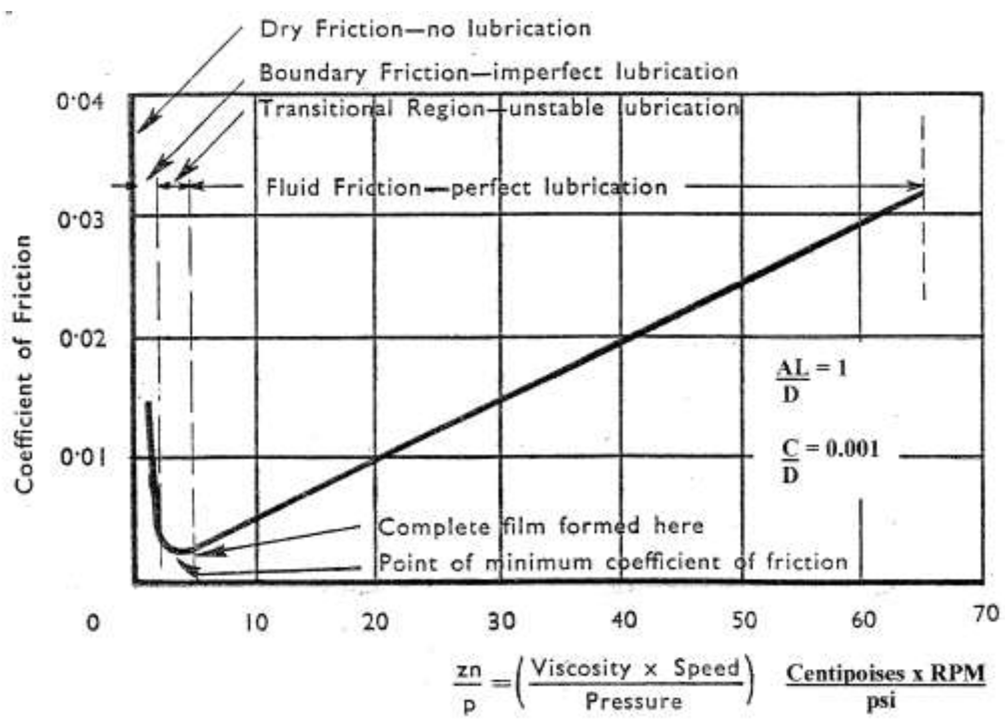
In a simple lubricated plain journal bearing the load-carrying capacity and the friction coefficient are determined by the non-dimensional parameter:-

$$\left(\frac{ZN}{P}\right)$$ (the full expression includes the Clearance/Diameter (C/D) ratio)

where:- Z = Viscosity of the oil at the temperature *inside* the bearing;
N = RPM;
p = Pressure, defined as:- Load carried per unit of projected Area, and Area = pin Diameter (D) x Axial Length (AL);

provided that $\left(\frac{ZN}{p}\right)$ is above a critical minimum,

after which the hydrodynamically-self-generated oil pressure *inside* the bearing carries the revolving pin clear of the static casing (625,626). The figure below from (625) illustrates the characteristic relation of Coefficient of Friction to $\left(\frac{ZN}{p}\right)$.



The above diagram is the "Stribeck curve" named after the German researcher who proposed it in 1902.

Since the internal oil pressure (whose components in the direction of the load integrate to 'p') has to be reacted on the bearing material the latter has to be capable of resisting it at the local oil temperature for the desired life. The characteristics of the material (and of the oil) are also of major importance during the start-up phase when $\left(\frac{ZN}{P}\right)$ is below critical. The lining's ability to absorb any grit in the oil without damage to the pin is also important.

Load and Speed variations

In a piston engine the cyclical loads on the "Bottom-End" bearings and the angular speed variation of the connecting-rod bearings add extra terms but these are favourable to load-carrying. This is seen in the simplicity of the highly-loaded Gudgeon Pin bearing, where the direction of rotation reverses during each crank revolution,: splash-lubricated hard bronze bushes supporting steel pins have remained scarcely changed (except to copper-lead material post-WW2) throughout the **h**istory of the internal combustion piston engine.

Frictional Power Loss

The frictional power loss in a plain lubricated bearing, running above the critical value of $\left(\frac{ZN}{P}\right)$ where the Coefficient of Friction is then approximately proportional to the parameter, is given by :-

Power Loss proportional to $Z \times V^2 \times AL$

where V = peripheral velocity of the pin

$\propto \pi \times N \times D$

(625,626)

Plain Bearing Axial Length/Diameter (AL/D) ratio

In order to reduce the oiled area and hence the frictional loss, the Axial Length/Diameter (AL/D) ratio of plain bearings has been reduced steadily with time as shown below:-

Date	Engine	Data Source	Average AL/D	Mean Piston Speed (MPS) m/s
1889	Daimler 14^0V2 0.6 L	SO 1	2.7	2.0
1895	Panhard IL2 1.2 L	SO 2	1.87	3.2
1908	Mercedes IL4 13.5 L	Eg. 3	1.59	9.6
1921	Duesenberg IL8 3 L	Eg. 7	0.94	16.6
1927	Miller IL8 1.5 L	SO 9	0.8	20.6
1932	Alfa Romeo IL8 2.65 L	Eg. 18	0.5	18.9
1936	Auto Union 45^0V16 6 L	Eg. 22	0.32 MJ only	14.2
1949	Ferrari 60^0V12 1.5 L	Eg. 27	0.46	13.1
1957	Maserati IL6 2.5 L	Eg. 35	0.47	18.8
1965	Coventry Climax 90^0V8 1.5 L	Eg. 42	0.55	16.3
1967	Cosworth FVA IL4 1.6 L	Note 79	0.43*	20.7
1992	Honda RA122E/B 75^0V12 3.5 L	SO 20	0.45	23.0
2002	BMW 90^0V10 3 L	(1095)	0.5	26.1

The reduction of AL/D in the Auto Union bearings, although with the 1st copper-lead lining (see later), proved to be a step too far for the crank pins, which may have been as low as 0.28, when the engine was enlarged to 6 L in 1936 (the original capacity was 4.4 L), probably because there was too much oil leakage. These bearings had to be converted to rollers (with solid races on a Hirth-type built-up crank).

After the introduction of "thinwall" linings in the post-WW2 Ferrari (see later) the average value of AL/D was generally around 0.5, as shown.

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*FVA: individual AL/D were:- MJ = 0.47; CP = 0.39. The Cosworth DFV AL/D for CP was the same 0.39.

1st Group

Whitemetal bearings

For the main journals and big ends of the 1st group of engines a very good bearing lining was available in a compound patented by Babbit in 1839, being Sn 89%/Sb 7%/Cu 4% or thereabouts

(625) and sometimes called by the patentee's name or alternatively "whitemetal". This would run against an unhardened steel shaft (170 Brinell) and was almost proof against acids produced in the oil during running of an internal combustion engine. It was capable of supporting around 12 MPa (≈ 1,700 psi) average pressure provided that surface temperature did not exceed 100C (610,626).

Whitemetal provided a sufficient racing life up to 1914, when the Mercedes IL4 4.5 L ran up to a "Red line" of 3,450RPM, equal to a Mean Piston Speed (MPS) of 19 m/s (468).

Early improvements in the 1st group

During the time period of the 1st group lubrication systems for plain bearings were developed from early "gravity + splash" arrangements (621)* into full pressure supply. The "dry", i.e. scavenged, sump with a separate oil tank appeared in the 1913 Peugeot (4).This provided cooling of the oil.

Crank design also advanced in this period from simple to counter-balanced to reduce journal loads

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*Although relating to a non-Grand Prix pure-sprint engine, (446) gives an interesting account of how in 1923 a customer specification "Brescia" IL4 1.5 L Bugatti with whitemetal bearings lubricated by a "Squirt-and-hope" system were made to stand an MPS of 20 m/s for a very short life by fitting oil pickup-scoops on the crank webs, a modification designed by Amherst Villiers.

Early bearing design

In the racing engines of the 1st group the whitemetal was used in a thick lining cast into thick bronze or steel shells (except the 1921 Duesenberg crank pin bearings which anticipated later "thinwall" bearings by having a reduced thickness of whitemetal cast directly into the tinned con-rod and its cap (711). This approach had been used previously by Henry Royce in the 1914-designed V12 20.3 L Eagle aero-engine. The Duesenberg apart, such bearings did not have the heat transfer advantages which were produced by the 1930 invention by Hopkins and Palm of the Cleveland Graphite bearing company of the "thinwall" steel shell with only 0.7mm whitemetal lining (625) – which was halved later(610). This is the type of bearing brought to the UK by G.A. "Tony" Vandervell in the '30s (68), of which more later.

The 1913 Peugeot exception to the 1st plain bearing group

There was a partial harbinger of the 2nd Roller & Ball bearing group and an exception to the 1st plain bearing group in the Ernest Henri-designed 1913 Peugeots. The IL4 5.6 L and 3 L engines for the Grand Prix de l'ACF and the Coupe de l'Auto races, respectively, had 3 ball races (one double) to carry the 2-piece bolted-up crank in a barrel case. The big-end bearings remained plain whitemetal.

Post-WW1 Henri engines and the Floating Bush

It is interesting to describe here the post-WW 1 1919 -1921 Henri engines which he designed for Ballot. The 1913 idea was extended to his new IL8 engines of 4.9 and 3 L for Indianapolis and the French GP. These had 5 ball races (2 double) on a 4-piece bolted-up crank. The plain big-end bearings were of a type which has drawn unfavourable comment (4,26), having a fully-floating bush, white-metalled on each side, between the rod and the pin. These comments overlook the high-speed performances of the engines at Indianapolis and in the 1921 inaugural Italian GP at Brescia (1st and 2nd), also the 2nd and 3rd places won in the 1922 Targa Florio by the identical design in the Ballot 2LS sports car (26).

It is also a fact that Harry Ricardo removed a big-end bearing RPM limitation on the Bristol Jupiter radial 9-cylinder 28.7 L aero engine by introducing a 1-piece floating bush, white-metalled on both side and perforated for oil passage, with a 1-piece master rod, a design retained on all Bristol engines until the early '40s (343,626,632). Henri's floating bushes were made in 2 parts fitted in bolted-up rods, which Ricardo tried but found inadequate but it seems that Ballot workmanship was good enough for its purpose.

Oil pre-WW1

Early engines used mineral oil. Castor oil was introduced by Wakefield in 1909 for aero engine use and had its first major racing engine success in 1911, when it was used in the Senior TT-winning Indian motor-cycle. The following year it lubricated the engines of the Coupe de l'Auto Sunbeams which finished 1st, 2nd and 3rd (1099). Thereafter it was used generally for many years. Castor oil, like other vegetable-base lubricants, has greater "oiliness" than mineral oil, a property identified by later research as shifting the transition point on the Stribeck curve to a lower value of (ZN/p) (594).

Consequently it protects bearings better in the start-up phase. It is also more adhesive than mineral oil.

<u>2nd Group</u>

Bugatti used all rolling bearings in his “works” IL4 1.5 L Voiturettes which triumphed at Brescia in 1921. At the same date, in the Grand Prix arena, FIAT fitted all roller bearings to their 1921 IL8 3 L , introducing split races and cages so as to retain a 1-piece crank. They continued these features in their subsequent racing car engines to 1927, (after which they concentrated on Schneider Trophy aero engines).

The Sunbeam engines of 1923 -1925 copied the FIAT bottom-end practice. Delage also followed FIAT’s lead in the 1923 60^0V12 2 L (allegedly this engine, with camshaft and auxiliary drives also on rolling bearings, had over 100 of that type (455)). Alfa Romeo, with ex-FIAT designer Vittorio Jano, kept the Turin firm’s bearing practice in his 1924 IL8 2 L P2, as did the 1926-27 IL8 1.5 L Delages, Bugatti over 1924-1931 also retained all roller & ball bearings in his IL8 2 L, 1.5 L and 2.3 L units.

<u>Plain bearing exceptions to the 2nd group</u>

While European engines were using rolling bearings, an American maker, Harry Miller with his IL8 1.5 L engines, was showing what *could* be done with plain whitemetal even when highly supercharged and running up to a Mean Piston Speed (MPS: MPSP at Peak Power) of 20 m/s, racing at high speed over 500 miles at Indianapolis (6).

The all-rolling-bearing engines, while of higher Mechanical Efficiency than plain bearings with AL/D around 1, must have been not only much heavier but also much more expensive in initial and running costs because of their numerous high-precision parts and short lives resulting from high contact stresses. Furthermore, a roller big-end, because of the variable peripheral speed, tends to have the rollers skid and incur flats if accelerated too rapidly when the oil is cold and thick (although they *do* avoid the low (ZN/p) problem of plain bearings). Therefore, in the shortage of money for motor-racing after the world depression reached Europe in 1930, the cheapness of plain whitemetal bearings led to their revival for Grand Prix racing in 1932-1934.

Jano, in particular, had built up his experience of plain bearings in the series of Alfa Romeo 6C and 8C sports cars beginning in 1925. He then used them in the pure racing Tipo B (or “P3”) Monoposto of 1932 with a value of AL/D reduced to 0.5, which would have reduced the frictional loss. He continued them in all his subsequent Grand Prix designs (except using rollers for the big ends of his enlarged 60^0 V12 4.5 L of 1937 (25)).

Maserati also used mainly plain bearings in their 1933 8CM, and they were Ferdinand Porsche’s choice for his 45^0V16 4.4 L P-wagen design which he sold to Auto Union in 1933.

<u>Copper-Lead bearing material</u>

Porsche did not seek ultimate RPM from his P-wagen engine and, as a private venture, probably kept its cost down with the idea of selling it to the new Auto Union group who were less well-endowed than Daimler-Benz, hence the plain bearings. However, he made use of a new material which had just been proven for aero engines. This was Copper-Lead (around 69% Cu, 30% Pb, 1% Sn) (usually called “Lead-Bronze” but Copper-Lead better reflects the low Sn content). It had been brought to successful practice in the mid-‘20s by Norman Gilman of the US Allison Engine Co. It consisted of a *mixture*, not a compound, of a Cu-Sn matrix filled with Pb, the latter smearing over the surface in start-up to provide the bearing surface. The basic problems, of keeping the material adhering to the still thick steel backing shell and of protecting the Pb from settling, had been solved by quenching after casting into the pre-heated shell. This initially-secret process was patented in 1926 (628). Charles Lindbergh’s TransAtlantic Wright J5 9-cylinder 13 L radial engine in 1927 was fitted specially with the new type of Allison bearing and gave it a great advertisement. Compared with whitemetal, both in “Thick-wall” form, Copper-Lead offered 25% higher pressure capability to a higher temperature (626). Being harder it required a pin of around 270 Brinell, better oil filtration because it could not embed grit very well and more accurate alignment since it could not distort to accommodate errors.

Having been bought by General Motors in 1929 Allison in 1932 developed for GM to produce an automotive steel-strip-backed Copper-Lead bearing (presumably following the Cleveland “Thin-wall” pattern of whitemetal type already mentioned). It may be that Porsche was the first European user of such bearings and, as Allison guarded their process well – e.g. Rolls-Royce had to buy a licence

and pay royalties to use it (628) – he must have paid for the privilege. The rather-daring reduction of AL/D in the P-wagen engine has already been mentioned.

Reversion to rolling bearings by Mercedes-Benz

When Mercedes-Benz re-entered the Grand Prix arena in 1934 with "cost-no-objection", and also true to their "advanced-but-conservative" policy, they used the split race and cage approach to employ roller bearings on a solid crank through to the time in 1939 that their country, once again, forced a halt to racing with another war.

Auto Union mixed bearings

It has already been described how Auto Union in 1936, when enlarging the engine to 6 L, were forced by the retained original 1934 crankcase dimensions to move from the first choice of plain bearings to rollers for the big ends.

Oil in the inter-War period

Castor oil was used generally in WW1 aero engines, both in rotary units and also in "stationary" engines. However, it had the unsatisfactory property of oxidising easily, leading to tenacious deposits. Therefore, in 1923 Harry Ricardo ran tests for the Air Ministry in an aero engine with a new Shell light mineral oil. This gave 10% more power than the previously-used castor oil, with "perfect" lubrication This was followed by tests in racing cars at Brooklands which produced 3% higher lap speeds, also consistent with + 10% of power (294). Thereafter racing had a choice of lubricants. Some people preferred to stay with castor oil because of the greater protection in the transition region.

Auto Union introduced an oil cooler into the scavenge line in 1936 (4), thought to be a 1st which became usual afterwards.

3rd Group

The return to plain bearings

The period of the 3rd Group in Grand Prix racing opened with the domination of the Alfa Romeo type 158-developed-to 159, which had been designed originally as a Voiturette by Gioachino Colombo in 1938. As the principal assistant to Jano during the'30s he had followed his former chief's later practice and adopted plain bearings for this engine and it is probable that these were Copper-Lead. By 1938 this material was in full aero use and was well-known for automobile use also, particularly in Diesel engines with their high peak bearing pressures. Backings were still thick-wall. The 158 needed something better for the big-ends, however, and needle rollers were introduced there, either originally or as an early modification, noting that Jano may have used them in his 1937 V12. Needle rollers are particularly suitable for the variable angular velocity of big ends but, being uncaged, they are certainly not "friction-less".

Enzo Ferrari originated post-WW2 the marque which would in 1951 defeat the Alfa 159 – although not over a full season – and go on to its unique place in Grand Prix (and all other) racing. Colombo designed for him a 60^{0}V12 1.5 L engine and, around 1946, ran trials to decide whether to use plain or roller bearings. The plain contender was now the Vandervell "Thinwall" 3-layer (Copper-Lead-Indium) type which had been developed during WW2 for the 1944 Mark V version of the Napier Sabre H24 36.7 L aero-engine (631,632). This bearing combined the Gilman basis, with Napier's introduction of an electro-deposited 0.001 inch (0.025mm) thick lead plating to cover the grit/debris start-up problems more effectively, plus a diffusion of Indium to protect the lead from acids produced in the oil during service, with the advantages of the thin shell. This thin shell and the very-thin layer of actual bearing material minimised fatigue due to cyclical internal stresses as the material heated and cooled – the major part of the cooling still being by oil flow, of course. In the new Ferrari V12 the AL/D ratio was just under 0.5. Statements were published that the Vandervell bearings had shown "+10% output" (633) – compared to the roller alternative, it is presumed. Even if this figure was optimistic, ref. (594) stated that roller & ball friction is not necessarily much less than a plain bearing in the full-oil-film region (which would be considering roller/ball frictional interaction at the cage versus a low AL/D journal together in each case with the necessary oil pumping + scavenging powers). At any rate, weight and lifetime cost also being considerations, thinwall-type bearings were chosen for the V12 Ferrari and then remained in all subsequent Ferrari engines (with one partial exception 30 years later, which will be described below).

Every consistently-successful Grand Prix engine adopted the same basic type of thinwall multi-layer plain bearing up to the present day, apart from the Mercedes-Benz M196 (see below).

Interesting information on the plain Vandervell Thinwall bearing life in the 1954 Maserati 250F IL6 2.5 L engine is given in ref. (147), as follows:-

RPM limit	MPS m/s	Life of bearings	Relative $(MPS)^2$	Relative Distance*
7,200	18	5 races run by Moss' car, before overhaul, say 1,900 km.	Datum	Datum
7,600	19	2 races advised by Maserati, Say 1,200 km.	+11.4%	-37%
7.800	19.5	1 race advised by Maserati, Say 600 km. The usual works limit.	+17.4%	-68%

*Assuming practice covered the equivalent of20% race distance. In those days an engine would be run for the whole event, including usually 2 days of practice (unless it failed) but would probably have the valves reground before starting in a major race (147).

Mercedes-Benz roller bearings – and a rethink

The Mercedes-Benz W196, with M196 engine, was the Champion car in 1954 and 1955 (and in adapted 3 L form for the 300SLR sports-racing car also the 1955 champion in that category). It was the major exception in the 3rd Group use of plain bearings. Mercedes had both a "conservative-pioneering" policy and a reliability target – often achieved - of *all* 3 or 4 team cars finishing and the rolling bearing engine can withstand an oil supply interruption for some time, which might occur due to unexpectedly-high consumption and low tank level near the race end.

Nevertheless, when it was found by bench tests that the 300SLR engine had to be restricted 7% below its peak power RPM if the roller bearings were to finish the 1955 24 hours of Le Mans* a mostly plain bearing crank was schemed for the intended-but-later-cancelled 1956 sports-racing programme (468).This life limit was despite the M196 basic design having been improved – at great expense – from the pre-WW2 Mercedes split-races/ 1-piece crank by changing to solid races/developed Hirth-type built-up crank.

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*In the 1955 Le Mans the Mercedes were actually withdrawn after 10 hours because of the 3rd team cars involvement in the spectator disaster of that year.

Ferrari mixed bearings

The Ferrari exception in the 3rd group of all- plain-bearing engines was the type 312B F12 3 L, Champion in 1975, 1977 and 1979. This began with 4 roller bearing main journals on a built-up crank but trouble with the latter led to a partial reversion with a solid crank having plain intermediates and the unsplit race rollers at each end.

Other non-"Car-of-the-Year" rolling bearing engines , rethinks and poor results

Porsche for their 1962 Grand Prix F8 1.5 L engine tried the same all-roller Hirth-crank pattern they had used in their previous racing engines but they settled on plain Copper-Lead bearings (635), obtaining a single win. This was, in effect, a re-run of the Ferrari tests of 1946.

Honda during 1965-1968, guided by Yoshio Nakamura, carried over their very-successful racing motor-cycle practice of roller & ball (R & B) bottom-ends to their 60^0V12 1.5 L and 90^0V12 3 L GP engines but had only isolated wins with each type. The 3 L was commented on as being relatively heavy. However, they did have great success with their 1966 IL4 1.0 L Formula 2 engine, also with all R & B.

In much more recent years, post the author's detailed review period but worth mentioning, the Peugeot 72^0V10 3 L GP engine of 2000 (at least) had roller bearing main journals, these having split races and a solid crank (1101, this describing the engine as taken over in 2001 by the Asia Motor Technologies company). The Peugeot GP engine programme, from 1994 to 2000, was not a success, the highest finishes totalling only four 2nd places in 7 years.

It was also reported (1102) that Toyota produced a development V10 GP engine with a *ceramic* roller main bearing crank which gained 20HP compared to plain bearings but the date was not identified. It was not raced due to regulation changes (probably the stepped life increases

demanded beginning in 2003) .The gain might represent rather over 2%. Whereas the weight penalty would have been less than all-steel the cost would no doubt have been far above normal rollers.

Plain bearing material development in the 3rd Group

Plain bearing materials have moved to harder specifications in the last 2 decades. Whereas the Coventry Climax engines of 1961 -1965 used Vandervell VP2, virtually the same chemical composition as the Air Ministry DTD229 of the '30s with 74% Cu, 23% Pb, 1.5% Sn + controlled impurities + Indium, the developed VP10 specification had 78% Cu, 10% Pb, 10% Sn. This latter material is more truly called "Lead-Bronze" and, being harder, requires corresponding improvements in the 3 areas of:- Pin Hardness (300 Brinell); Oil Cleanliness; and Shaft Alignment. It can run to an average p = 82 MPa (12,000 psi), a 7-fold improvement over the century, and to a higher oil temperature: "above 125C" was quoted in 1997 (567). However, prior to 2003 the bearing life required would have been only 1 race + warm-up, about 400 km, an overhauled engine having been fitted after Qualification.

Crank diameter reduction

With bearing AL/D already reduced to a minimum, further advantage was taken in the '90s to reduce pin diameters as a proportion of stroke, even although RPMs had risen. This produces a square-law reduction in frictional power loss, as shown above. Vee 10 cranks were so "skinny" that they ran through a critical period at about 70% of peak power RPM (1095,255) – with electronic engine control systems a dwell on the critical speed could be avoided. That designers *were* prepared to take some risks to minimise journal (Diameter x Length) so as to reduce friction was illustrated in 1998 by 2 race retirements in 1998 of the Championship-winning Ilmor-Mercedes-Benz FO110G engine because of admitted main bearing failures.

A later report (1095) shows that BMW reduced their main journal and crank pin diameters for the 2002 season's P82 engine despite raised Red-line RPM, although it does not reveal by how much.

Oil in the post-WW2 period

Until about mid 1970 post-WW2 engines were lubricated with mineral oil modified with various additives to improve film strength and resist higher temperatures. A very-significant exception was the Mercedes-Benz M196 all-roller-bearing engine, where it was found that the latest Castrol R oil was the most suitable lubricant (468) (but this choice *may* have been forced to cope with the desmodromic valve gear).

In the early '70s oil made indirectly from crude petroleum was developed, derived from the chemical family of poly-alpha olefins (PAO) and generally called "synthetic". This had further significant quality advantages which enabled engines to be raced with less oil carried and smaller de-aeration and cooling systems (1100).

Some 15 years later another type of synthetic oil was available, originally developed for aero gas turbines, based on poly-ol esters. This reduced friction, hardly suffered aeration and withstood very high temperatures. It solved the problem of lubricating the highly turbocharged engines of 1983 - 1987 (1100). Mobil was supplier of this "dream oil" to Honda.

Some indication of continuing improvement in racing oils is given by 2 reports:-

- For the Italian GP in September 1993 Ayrton Senna's race engineer, Georgio Ascanelli, advised him that his Cosworth HB8 engine would benefit by 0.4% of power from a new oil (636);
- Mario Theissen stated in (1095) that over the 10 years of development of BMW F1 engines from 2000 to 2009 oil development had contributed + 4½% of power.

Note 18

Grand Prix engines Main Journal (MJ) and Crank-Pin (CP) Bearings (1): 1906 – 2000

Data sources as in Appendix 1. Notes (1) to (9) on next page, also *, ** and ***.

YEAR	ENGINE	CN (2)	MJN (2)	BEARING TYPE (3)				
				PLAIN			ROLLING (4)	
				WHITEMETAL (5)	COPPER-LEAD (6)	AL/D (7)	CRANK TYPE	
							SOLID,	BUILT-UP,
							SPLIT RACES (8)	SOLID RACES (9)
1906	Renault	4	3	● (10)				
1908	Mercedes	4	3	● (10)		1.59		
1911	FIAT	4	3	● (10)				
1912	Peugeot	4	5	● (10)				
1913	"	4	3	● CP				● MJ (B)
1914	Mercedes	4	5	●				
WW1								
1921	Duesenberg	8	3	● 2 MJ ● CP		0.94	● Rear MJ	
1922	FIAT	6	8				●	
1923	Sunbeam	6	8				●	
1924	Alfa Romeo	8	10				●	
1925	Delage	V12	7				●	
1926-27	"	8	9				●	
1928-31	Bugatti	8	5					● 3 MJ ●CP(C)*
1932-35	Alfa Romeo	8	10	●		0.5		**
1933	Maserati	8	5	● 3 Inter MJ ● CP			● 2 End MJ	
1935-36	Mercedes	8	5				●	
1934	Auto Union	V16	10		● (10)	0.32		
1935-36	"	V16	10		● MJ	"		● (CP) (H)
1937	Mercedes	8	9				●	
1938-39	"	V12	7				●	
WW2								
1947-51	Alfa Romeo	8	9		● MJ		● CP Needle Rollers	
All plain bearings were "thinwall" copper-lead from this date								
1949	Ferrari	V12	7		●	0.46		
1952-53	"	4	5		●	0.43		
1954-55	Mercedes	8	10					● (H)
1956 Onwards	All CoY except Ferrari 1975, 1977, 1979		CN per bank + 1		●	0.4 to 0.5		
1975,etc	Ferrari	F12	4		● CP ● 2 MJ		● 2 MJ***	

Notes to Table on P.8.

(1). Nearly all Gudgeon-Pins (GP) were steel running in plain hard bronze bearings, splash-lubricated, except that 1934 – 1937 Auto Unions had Needle Roller bearings, as did the 1954-1955 Mercedes-Benz. In engines post-WW2 GP bearings were copper-lead and since 1958 the lubrication will have been enhanced by crank-case oil jets used to cool the piston crowns.

(2). CN = Number of cylinders; MJN = Number of main Journals.

(3). Bearing type is same for MJ and CP unless shown otherwise.

(4). Usually Rollers, with a Ball-bearing for crank axial location.

(5). Bearings cast into thick bronze or steel shells except Duesenberg CP where whitemetal was cast directly into rods.

(6). "Thinwall" type beginning with Ferrari in 1947.

(7). AL = Axial Length; D = Diameter.

(8). Except those rolling bearings which could be threaded complete onto crank ends.

(9). Types of Built-Up cranks:-

- (B) = Bolted flanges;
- (C) = Taper and Key and/or Cotter-Pinned;
- (H) = Hirth proprietary type, with dogged faces and bolts. Post-WW2 with differentially-threaded bolts.

(10).These engines had cranks with no counter-weighting. Others counter-weighted.

*Bugatti: + 2 intermediate MJ which had split races with crowded rollers to avoid split cages.

**Alfa Romeo: 2 x 4 cylinder cranks bolted together between the 2 centre plain bearings.

***The Ferrari 312B had a roller bearing at each end of a solid crank and 2 intermediate plain main bearings.

Note 19

Other Mechanical and Thermal Limits

In addition to the Mechanical limits already listed in Note 13, other limits were (and are):-

Instantaneous in destructive effect :-

Valve motion out-of-control

With OHV engines, (excluding Desmodromic valve gear, see below) as Compression Ratio (R) was raised and Valve-Opening-Periods extended, there was created the certainty that, above a critical RPM, an exhaust valve would fail to follow the cam home and it would be overtaken by the rising piston. The actual critical RPM would depend on the type of valve gear and the quality of the cam contour.

At the least the valve would be bent and so power lost; at the worst a valve head would be broken off and/or a piston holed and the engine could be destroyed

For many years the only safeguard was a "Red Line" on the tachometer and a trust in the driver to obey it – which often was ignored in the heat of battle (see also Sub-Note A). In 1938 Mercedes-Benz experimented with an automatic rev-limiter but did not find it sufficiently reliable to use in races (468)(see also Note 82). For the Cosworth DFV in 1967 a reasonably-reliable rev-limiter was available and this, gradually improved, became a standard for Grand Prix engines (see also the descriptions of DFV development in "The Unique Cosworth Story". It protected the engine against missed upward gear changes or deliberate over-speeding but could not guard it from premature downward changes when the car's speed would over-rev the engine.

With the arrival in 1989 of electronic engine management systems linked to semi-automatic gear-change-cum-clutch operation mechanisms,initially by Ferrari, the over-revving problem was solved completely (see Sub-Note B).

Desmodromic valve gear – mechanically-controlled valve motion –could also solve the problem, of course, but the cost of doing this successfully was only tolerable by Mercedes-Benz in the 1954-1955 M196. No other CoY engine in this review used it.

Time-related in destructive effect:-

Crankshaft torsional vibrations

See the description of the Mercedes-Benz M196 (Eg. 32) (2NA page 4).

Camshaft-drive vibrations

See the descriptions of the Cosworth DFV development "The Unique Cosworth Story" and also of the Coventry Climax 4 valve per cylinder engine (Eg. 44)(2 NA page9).

Thermal limit to RPM

An example of an instantaneous *Thermal* limit to RPM is given in Sub-Note C.

Sub-Note A

Avoiding over-revving

Ricardo limited the 1922 3 L TT Vauxhall engine to R = 5.8 so that valve-piston clash could not occur if a valve stuck open or if the engine was over-revved until the valve left its cam (4). The Bugatti vertical-valve engines (i.e. those preceding the 1931 type 50 et seq "Miller-head" engines) could run very high R (up to 13.5 in a type 37A (308)) without risking a valve-piston collision. Although not run that way for GPs it made them very popular with British tuners for sprint races and hill climbs.

On the other hand, the 1922 FIAT 2-valve/cyl. head with VIA = 102^0 was probably the first engine at risk if over-revved. The normal practice in later years as R was raised was to provide cut-outs in the piston crown to reduce the likelihood of contact. The Mercedes-Benz 4-valve/cyl. engines of the 1934 – 1939 period were given raised "buttons" on the piston cut-outs, on the valve centre-line, in the hope that any contact would not bend the valve.

Certainly *some* over-revving past the normal race-life limit was possible in these Mercedes if needed for a last-lap overtaking opportunity against the Auto Union 1934 – 1937 V16s, whose push-rod actuation of the exhaust valves could not accept emergency over-rev without failure (607).

Von Brauchitsch of the pre-WW2 Mercedes team was a notorious frequent over-revver and the works sought to control him by an over-stated tachometer dial. These experiences were part of the reason for adopting mechanically-closed valves for the 1954 – 1955 M196 engines (468).

Another way of trying to control drivers had been tried on the 1927 Delage by having coloured segments on the very-large tacho dial – green from 6,000 to 7,500 RPM, then yellow to 8,000 and red from 8,000 to 9,000. Another simple practical step was to mount the tacho with the max. recommended RPM at the top so that a quick glance at a vertical needle was re-assuring. On the Delage the dial-top number was 7,500 RPM (1075). The 1936 Auto Union, with its noted sensitivity to over-revving, had a red sector on the tacho from 3,500 to 5,000 RPM (381).

It is interesting to note that the first piece of "stored-data-readout" from a GP engine was the max.-RPM-reading non-return needle on the tachometer, introduced post-WW2. Now this has been long-superseded by multi-channel recorders radioing data back to the pits.

Sub-Note B

In the 2003 Cosworth CR5 V10 3 L, taking advantage of the electronic RPM control, the clearance between the valves and pistons was so small that no carbon formed on the piston crown between those parts (883).

Sub-Note C

Thermal limit

Apart from time-related thermal limits to the life of pistons, described in Note 14, a case is known of an instantaneous thermal limit to RPM of the 1936 MG type Q/Ex127 IL4 750 cc record engine. On the test-bed this highly-mechanically-supercharged engine ran into a backfire problem. Sydney Enever determined that this was pre-ignition from over-heated exhaust valves, which were solid KE965. He lengthened the valve guides to improve heat flow out of the valve stems by cutting the bronze guides and inserting extra cast-iron sections machined from available bar stock. This solved the problem (139). However, the engine (at 7,500 RPM) was still "Rated" below its 'natural' peak power.

An *apparently* similar problem in 1953 with the developed BRM T15 V16 1.5 L – backfiring on the test-bed at 11,500 RPM – was in that case put down by Tony Rudd to the back-pressure from inadequate rig silencers (40). In the car the engine could rev to 12,000 – an inversion of the usual situation for all power plants where the installed performance does not reach the test-bed figure!

To return to home page: Home Page

Note 20

Coventry Climax: Specific Power variation with Stroke

The 1953 – 1966 Coventry Climax series of DOHC Naturally-Aspirated racing engines provides a good illustration of the effect on Volume Specific Power (PP/V) and Piston Area Specific Power (PP/PA) of shortening Stroke (S). The details and charts are given on pp.2, 3 and 4.

This series has the valuable advantage for comparison purposes of a common design, detailing, manufacturing and development philosophy under one chief, Walter Hassan, and all powers tested to the same standard have been published by the company.

To remove the (fairly small) variation of Compression Ratio (R) the powers have been "normalised" by Air Standard Efficiency (ASE) to R = 12 and, for the 2 units running on alcohol-base fuel, an adjustment to petrol equivalent of1/1.12 has been applied (these adjustments are RA and AA respectively, as described in the Key to Abbreviations to Appendix 1). This "normalised" Peak Power is identified as PPA. Purely to give a convenient number 1/S is shown as (100/Smm).

It can be seen on Fig. 104/DST on p.2 that for this engine series – in which the 1st, the V8 FPE was not fully-developed – there is a steady decline in the gain of PPA/V as 100/Smm increases. The last engine produced in 1965 for the 1.5 L formula, the F16 FWMW, may have been very near the maximum attainable, although it also was not fully developed before the project was dropped.

In choosing 16-cylinders in late 1963 for an engine which he hoped would be available for racing in 1965, Hassan was unfortunately influenced by the old theory:-

"Power proportional to Piston Area".

This of course *would* be true if Brake Mean Effective Pressure (BMEP) and Mean Piston Speed (MPS) were constant. He took the PP/PA achieved by the then-current FWMV3 at 4.5 HP/sq. in. (=0.7 HP/cm^2) and thought that an increase in PA of 27% (45 to 57 sq.in.) would provide an engine giving 240 to 250 HP (34). See Fig. N20A on P.4.

Fig. 105/DST shows clearly that the "old theory" does *not* apply even for engines of given technology. The FWMW never gave over 209 HP.

Fig.106/DST on p.3 illustrates the way that MPSP (MPS at Peak Power) drooped off in the Climax range by about 25% as 1/Stroke was reduced to raise RPM. Apart from "normal" friction losses the F16 seems to have had an oil churning problem to reduce its Mechanical Efficiency (EM).

Fig. 107/DST suggests that, with an *average* BMPA of 12.1 Bar, there was an optimum around 12½ Bar (+ 3%) in the middle of the range. The FWMW, with the smallest cylinders of the series (B = 54.1 mm; S = 40.6 mm) had actually fallen off to only 10.4 Bar as flow pressure loss and heat loss increased, i.e. Volumetric Efficiency (EV) and Combustion Efficiency (EC), as well as EM, fell, as

(Surface Area/ Volume) proportional to (1/Characteristic Dimension) increased.

Fig.108/DST gives BMPA translated into Combined Efficiency, ECOM%, for these Naturally-Aspirated engines for which Manifold Density Ratio (MDR) was 1, so

$$ECOM\% = BMPA \times 100/23.94.$$

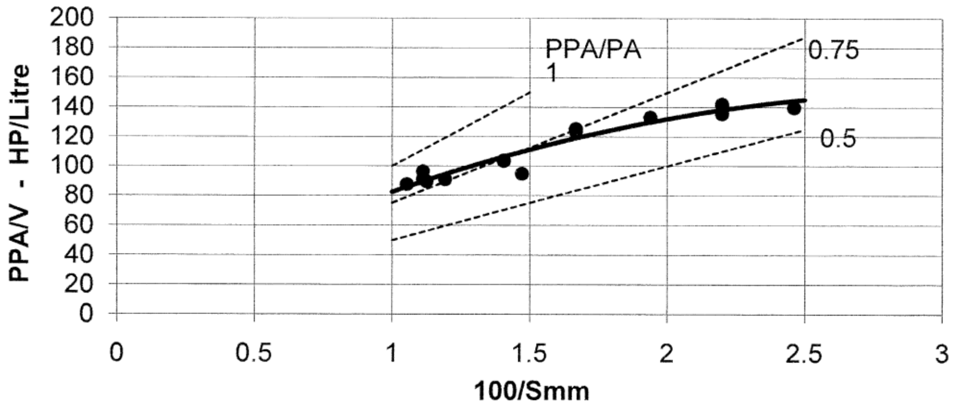
The average value was 50.7%. The best value was for the 90V8 FWMV Mk 1 1.5 L at 55.2%. As that family of V8 engines was developed to shorter Stroke and higher power, ECOM declined to 50.7% at the Mk 7. The F16 FWMW 1.5 L was only 43.6%.

COVENTRY CLIMAX. DOHC Racing Engines. Normally-Aspirated

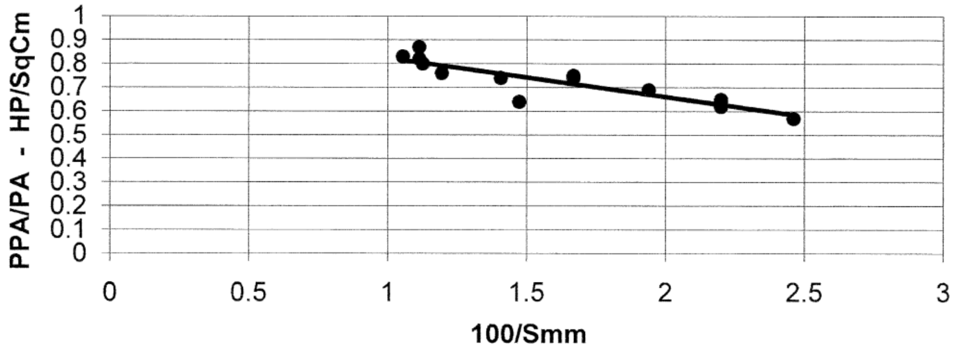
Sources: DASO 33, 34, 54, 57, 131, 131B, 249, 515.

S/No.	Year	Type	V cc	100/Smm	PP HP	R	RA	AA	PPA HPnormal	PPA/V HP/Litre	PPA/PA HP/SqCm
1	1953	FPE	2479	1.472	258	11	1.021	1.12	235.2	94.9	0.64
2	1957	FPF	1476	1.406	146	10	1.047	1	152.8	103.5	0.74
3	1957.5	FPF Mod	1964	1.193	180	12.4	0.993	1	178.7	91	0.76
4	1958	FPF Mod	2207	1.125	194	11.1	1.019	1	197.7	89.6	0.8
5	1959	FPF GP	2496	1.112	220	10.3	1.038	1	228.5	91.5	0.82
6	1960	FPF GP	2496	1.112	240	11.9	1.002	1	240.5	96.4	0.87
7	1961	FPF Mk2	1495	1.406	151	10.7	1.029	1	155.3	103.9	0.74
8	1961.5	FPF Indy	2751	1.053	270	12	1	1.12	241.1	87.6	0.83
9	1961.8	FWMV 1	1495	1.668	181	10.4	1.036	1	187.5	125.5	0.75
10	1963	FWMV 3	1496	1.939	195	11	1.021	1	199.2	133.2	0.69
11	1964	FWMV 5	1497	2.199	203	12	1	1	203	135.6	0.62
12	1965	FWMV 6	1497	2.199	212	12	1	1	212	141.6	0.64
13	1965.1	FWMV 7	1497	2.199	213	12	1	1	213	142.3	0.65
14	1965.2	FWMW	1495	2.461	209	12	1	1	209	139.8	0.57
15	1966	FWMV 10	1974	1.668	244	12	1	1	244	123.6	0.74

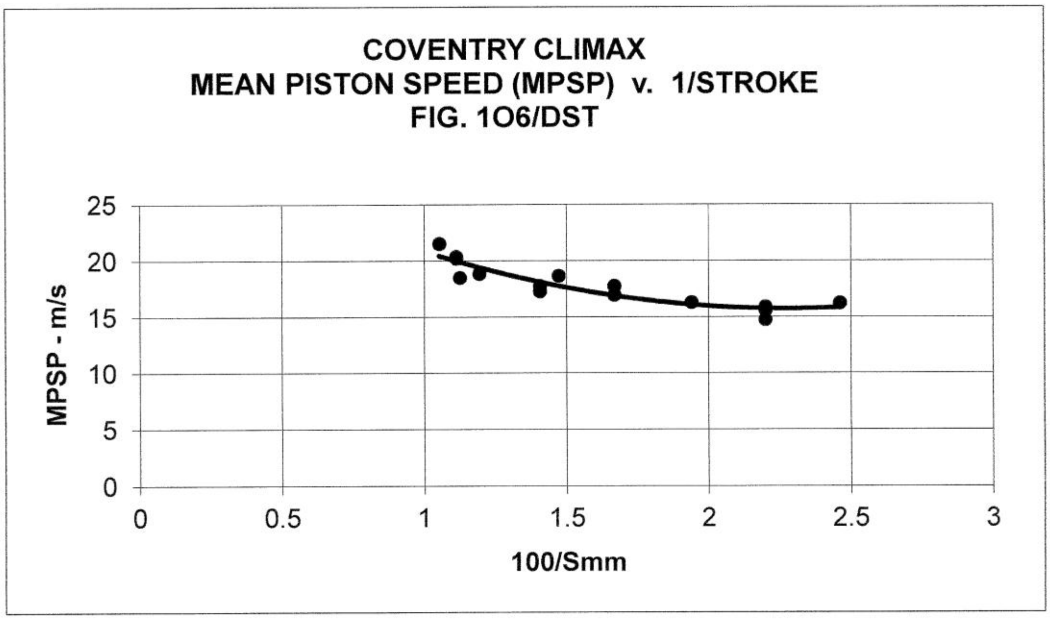
COVENTRY CLIMAX
POWER/SWEPT VOLUME v. 1/STROKE
FIG. 104/DST

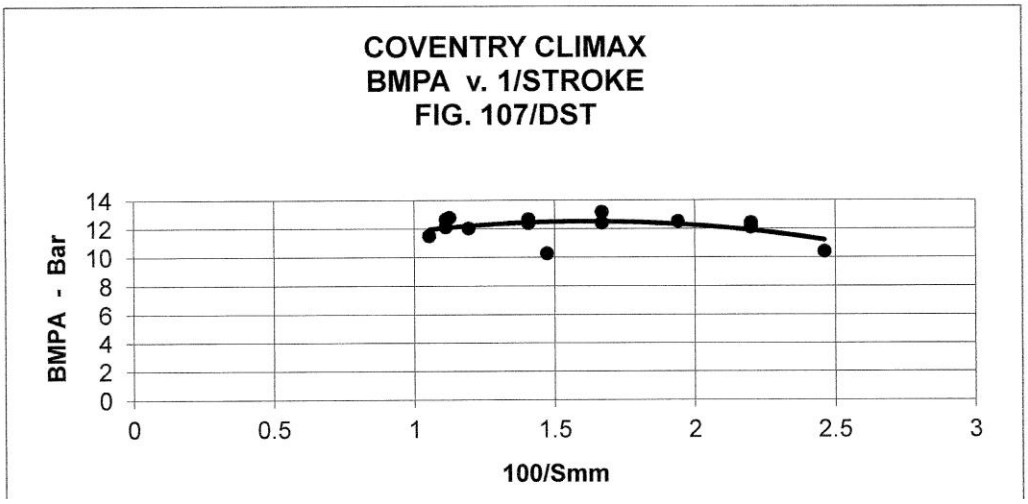


COVENTRY CLIMAX
POWER/PISTON AREA v. 1/STROKE
FIG. 105/DST



S/No.	Year	Type	V cc	100/Smm	ECOM%	MPSP m/s	BMPA Bar
1	1953	FPE	2479	1.472	43.0	18.68	10.29
2	1957	FPF	1476	1.406	53.0	17.31	12.69
3	1957.5	FPF Mod	1964	1.193	50.4	18.86	12.06
4	1958	FPF Mod	2207	1.125	53.6	18.52	12.82
5	1959	FPF GP	2496	1.112	50.7	20.23	12.14
6	1960	FPF GP	2496	1.112	53.0	20.38	12.68
7	1961	FPF Mk2	1495	1.406	51.8	17.78	12.4
8	1961.5	FPF Indy	2751	1.053	48.2	21.53	11.53
9	1961.8	FWMV 1	1495	1.668	55.2	16.98	13.21
10	1963	FWMV 3	1496	1.939	52.4	16.33	12.54
11	1964	FWMV 5	1497	2.199	52.0	14.78	12.45
12	1965	FWMV 6	1497	2.199	51.4	15.61	12.31
13	1965.1	FWMV 7	1497	2.199	50.7	15.91	12.13
14	1965.2	FWMW	1495	2.461	43.6	16.26	10.43
15	1966	FWMV 10	1974	1.668	51.9	17.78	12.43





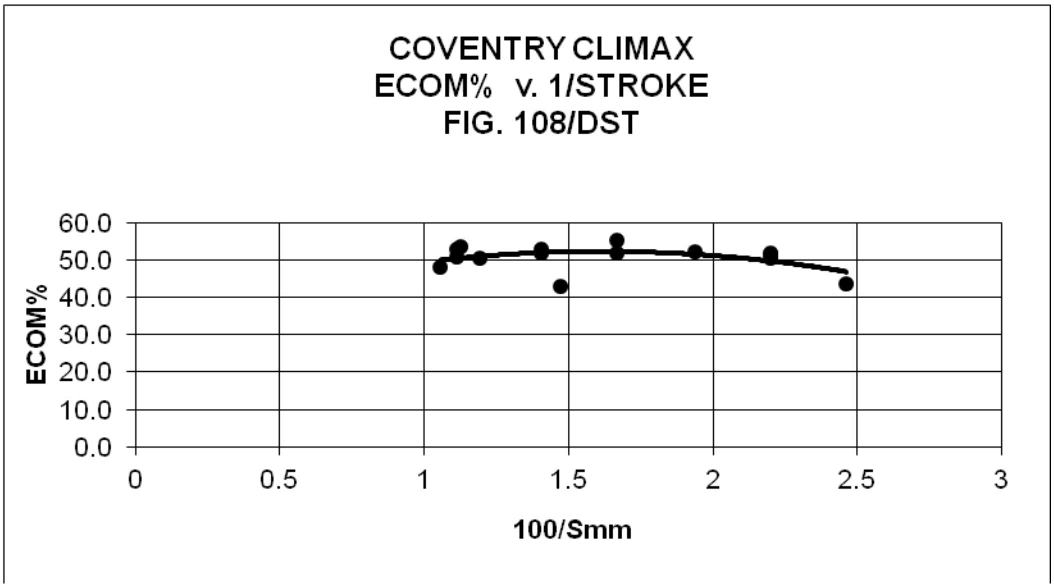
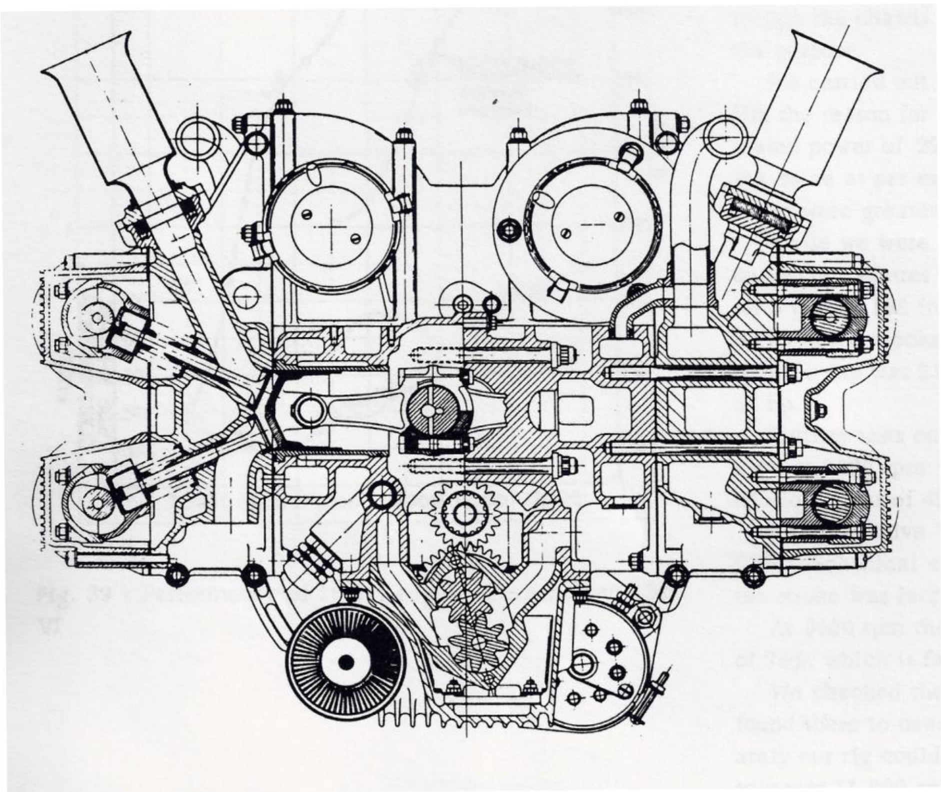


Fig. N20A
1965 Coventry Climax FWMW
F16 2.13''/1.60'' = 1.331 91.22 cid
(54.102mm/40.64 1,495 cc)



Note 21
Optimum Bore / Stroke ratio

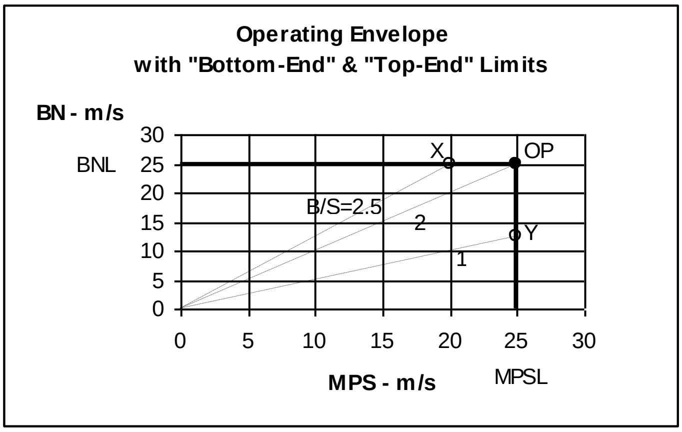
When the “Bottom-End” and “Top-End” limits of a poppet-valve piston engine are set by **MPS** and **BN,** respectively, according to the design conventions and available materials at a particular date and when **BMEP** is at the best attainable level with the knowledge at that date and in accordance with regulations, then for that case:-

$$\frac{\mathbf{P}}{\mathbf{V}} \propto \mathbf{N.}$$

There will be an optimum **B/S** ratio which maximises **N** and therefore **P/V**.

This is shown in the following analysis, which is for engines of fixed **V** and fixed Number of Cylinders (**CN**).

Let (**Limiting MPS**) be **MPSL** and (**Limiting BN**) be **BNL**. A diagram of the Operating Envelope will show (with limits typical of the later review period):-



The slope of the line from the origin to any point is equal to ½.(**B/S**) for that point.

For an Engine operating at point X

Since BN = BNL and MPS at X is < MPSL, therefore B/S at point X is *higher* than that at the limits intersection point OP.

Suppose it is required to alter the engine to raise MPS by a factor “f”. As BNL = ½.(B/S).MPS, therefore (B/S) must be *reduced* by a factor 1/f. Because V and CN are fixed, (B^2.S) is fixed and so (B/S) $\propto$ B^3 and the reduction ratio required of B = $1/(f)^{1/3}$. Since BNL is fixed, this reduction of B allows an $(f)^{1/3}$ rise in N. For e.g., a 10% increase in MPS is obtained with a 3.2% rise in N.

Thus *moving point X towards point OP creates a rise of N.*

For an Engine operating at point Y

Since MPS = MPSL and BN at Y is < BNL, therefore B/S at point Y is *lower* than at point OP.

Suppose it is required to alter the engine to raise BN by a factor “f”. As MPSL = 2.BN/(B/S), therefore (B/S) must be *raised* by a factor f. As before, (B^2.S) is fixed and so (B/S) $\propto$ $(1/S)^{3/2}$ and the reduction ratio required of S = $(1/f)^{2/3}$. Since MPSL is fixed, this reduction of S allows an $(f)^{2/3}$ rise in N. For e.g., a 10% rise in BN is obtained with a 6.6% rise in N.

Thus *moving point Y towards point OP creates a rise of N.*

Conclusion

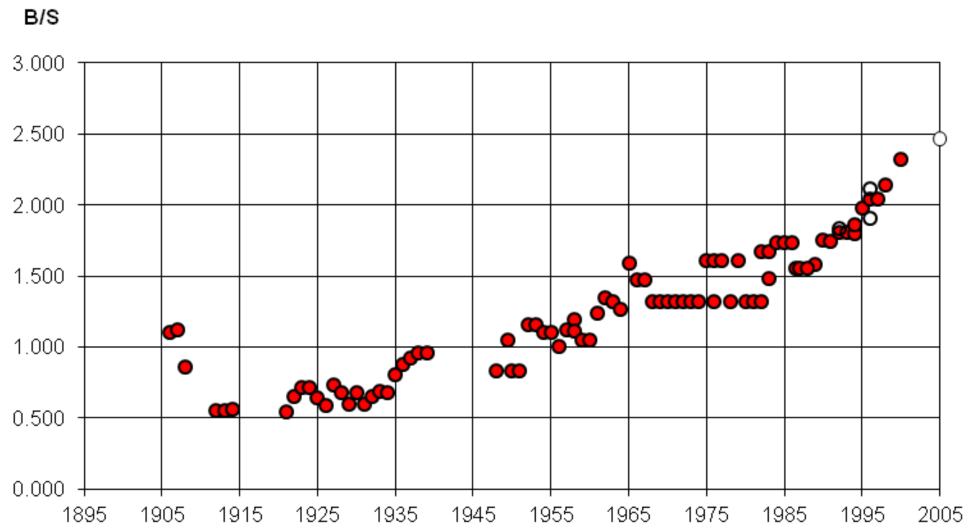
It follows from the above that the intersection of the “Bottom-End” and “Top-End” limits, point OP, represents maximised N and therefore maximised P/V for the chosen value of CN. The value of B/S at OP is optimum **for the design conventions and materials of the particular date**, at:-

$$\mathbf{(B/S)OP \ = \ 2.(BNL)/(MPSL)}$$

Historical Bore/Stroke ratio

The historical B/S ratio for CoY engines from 1906 to 2000 is given on Fig. G2 on P.2.

FIG. G5 BORE / STROKE



This chart has been extended to 2005 to include the BMW P85 V10 3L engine, which was the highest B/S ratio reached up to the time of writing (98mm/39.75 = 2.465). The unit was not raced because of a late change to the rules by FIA demanding longer life for which BMW had not designed. The engine was also the last in which the designer had a free choice of maximum B/S ratio, although constrained by a 2001 rule to 10 cylinders. After 2005, for the 2006 – 2013 V8 2.4L formula a maximum Bore of 98 mm was imposed, meaning a generally adopted B/S ratio like the P85.

For the new 3rd PC Era formula beginning in 2014 a maximum Bore of 80 mm has been regulated in V6 engines of 1.6L so that the highest permitted B/S is about 80/53 = 1.51.

It is now unlikely that the B/S ratio will ever be permitted to exceed the P85 figure.

Two comments on the historical series:-

- The initial drop of B/S to around 0.5 is explained in Note 35 "The influence of Maurice Sizaire on piston engine design";
- The long series of B/S around 1.3 from 1968 to 1982 is due to the continuing success of the Cosworth DFV (see "The Unique Cosworth Story").

The valve-gear developments which enabled the B/S ratio to increase so as to raise the volume-specific power are described in Note 15 "Valve-spring problems and their solutions".

Note 22

Design Parameters for other engines

All non-CoY Grand Prix engines

Figure numbers as in Design Parameter section

G4	G5	G6	G8	G9	G10	G12	G14
100/Smm	B/S	R	VIA^0	IVA/PA	IVL/IVD	CRL/S	B/PH

Plotted as O

1992 Honda RA122E/B [SO20 in Appendix 1] DASO (69, 711)

75V12; B = 88 mm; S = 47.9 mm; V = 3,496 cc

G4	G5	G6	G8	G9	G10	G12	G14
2.088	1.837	12.9	29	0.344	0.315	2.32	1.96

1996 Yamaha OX11A (Judd JV) DASO (674)

72V10; B = 90 mm; S = 47.13 mm; V = 2,998 cc.

G4	G5	G6	G8	G9	G10	G12	G14
2.122	1.91		25	0.348			2.14

1996 Mugen Honda MF301 DASO (672)

72V10; B = 93 mm; S = 44.1 mm; V = 2,996 cc.

G4	G5	G6	G8	G9	G10	G12	G14
2.267	2.11		23	0.336		2.72*	2.3

*The CRL/S ratio was unusually high for the time because the engine had been designed as 3.5 Litres and when the regulation swept volume was reduced to 3 Litres for 1995 at short notice (following the death of Ayrton Senna) the unit had been reduced with a short-stroke crank in the original cylinder block, necessitating a lengthened rod.

2005 BMW P85 Prototype DASO (1095; *Ten Years of BMW F1 Engines*. Paper by Prof. Dr-Ing. Mario Theissen et al. 2010)

90V10; B = 98 mm; S = 39.75 mm; V = 2,998.5 cc.

G4	G5	G6	G8	G9	G10	G12	G14
2.516	2.465			0.359			

Not plotted

2006 Cosworth CA/6 DASO (1069; *Race Engine Technology* No.20, Feb 2007). DASO (1107; *Race Engine Technology* No.73, Sept./Oct. 2013).

90V8; B = 98 mm; S = 39.77 mm; V = 2,399.9 cc.

G4	G5	G6	G8	G9	G10	G12	G14
2.514	2.464	13.3	18	0.355	0.387	2.573	

+ 6^0 in the plane of the valve pairs (longitudinally)

2009 Toyota RVX-09H DASO (1091; *Race Engine Technology* No.49, Sept/Oct 2010)

90V8; B = 96.8 mm; S = 40.75 mm; V = 2,399 cc

G4	G5	G6	G8	G9	G10	G12	G14
2.454	2.375	13.6	21.2	0.359	0.376	2.724	

+ 3.2^0 in the plane of the valve pairs (longitudinally)

Note 23

Fuel, Combustion Chambers and Compression Ratio

Early engines

Gottlieb Daimler's Maybach-designed 1st engine for automotive use in 1885 had a Compression Ratio (R) of about 2.4. This was with a combustion chamber extending beyond the side of the cylinder to accommodate a side exhaust valve and suction-operated overhead inlet valve. The 1895 Panhard with a similar chamber layout, also Maybach designed, was about R = 3 (2).

By 1906 the side-valve Renault French Grand Prix winner (Eg. 1) had R = 4. Petrol at that date, by the grading system created by Edgar in 1927 applied retrospectively, was about 45 or 50 Octane Number (ON) (apparently equivalent to the later "Motor Method" ON, see Note 58-2).It is noteworthy that in their contemporaneous writings on engines neither Frederick Lanchester nor Dugald Clark mentioned compression ratio, as though expecting it to remain at the levels then current.

However, by the movement in 1907 to more compact over-head-valve combustion chambers, it was possible in the years just before WW1 for R to reach about 5½; in 1914 for the Mercedes French Grand Prix winner (Eg. 6) fuel was improved by mixing 50% Benzole with 50% petrol (468).

Ricardo's fuel discoveries

During the latter part of WW1 Harry Ricardo discovered that petrol refined from Borneo crude oil could add a unit to the value of R typical with the then-best standard (probably Persian-oil-based) fuel. The 1st non-stop direct Atlantic crossing by air in 1919 was with the Rolls-Royce *Eagle* engines taking advantage of this better petrol to reduce fuel consumption (343).

Post-WW1 GP engines took advantage also of better fuel as well as Al-alloy pistons and smaller bores and the 1922 FIAT IL6 2L (Eg.8), with a near-hemispherical 2-valve combustion chamber, reached R = 7. The 1923 Sunbeam (Eg. 9), closely derived from the FIAT, had R = 7.4 (Note 32 discusses the post French GP strip condition of this engine).
This was the last CoY to burn plain petrol without additives.

Meanwhile Ricardo had also pioneered the use of Ethyl-Alcohol-base fuel which not-only enabled R to be raised over the best petrol but, by cooling the inlet charge much more than petrol by its greater latent heat of evaporation, produced a greater mass of air inhaled. Thus power was increased in two ways *but* Specific Fuel Consumption was doubled. This new "RD" (Ricardo-Distillers) fuel was used in the early '20s only for short-distance Brooklands races and hill-climb specials, e.g. Raymond Mays' Bugattis especially (446), probably because in such events the extra fuel consumption did not matter.

Alcohol-base fuel in Grand Prix racing

When Pressure-Charging (PC) was adopted generally after 1923, alcohol-base fuel was soon accepted as the way to avoid detonation by cooling the compressed charge. The value of R was restricted for the next 27 years to between 6 and 7 while Inlet Valve Pressure (IVP) rose from 1.5 to 3.9 ATA and the mixtures grew steadily richer in alcohol. Petrol and Benzole, the other major constituents of racing fuel, were steadily reduced (see Appendix 2) and the fuel/air ratio also grew richer relative to stoichiometric (chemically-correct) ratio. Racing miles-per-gallon therefore fell steadily and two replenishment stops were needed in 500 km by 1951.

Alcohol –base fuel remained the staple diet after the end of the 1st PC Era, into the 2nd NA Era, from 1952 to 1957 with R from 12 to 13. Nitro-Methane, an oxygen-bearing liquid, made its appearance as a fuel additive towards the end of these years.

The change to petrol

The companies who supplied free fuel for racing then forced a change to the rules to require "Pump petrol" with the aim of improving their advertising link from GP winners to "the petrol the ordinary motorist can buy!". However, difficulty at first in choosing an international standard

led to the specification of Genera Aviation petrol of 100/130 grade (see Note 58-2) for 1958 to 1960. Of course, this was *nothing* like petrol on sale to motorists!
Useable R was nearly as high as on Alcohol but the evaporative cooling gain was lost, e.g. the 1958 Vanwall had R = 11.5 and power output was down (after unrelated improvements) by about 4% (68).

Squish and Swirl

"Squish" and "Swirl" were both factors in obtaining high values of R in the Vanwall as direct contributions from its origin in the 1952 Norton motorcycle racing engine. "Axial Swirl" of the inlet charge inside the cylinder was induced by curving the port in the plane of the bore. The rising piston then compressed the swirl into the smaller diameter of the combustion chamber and so accelerated its angular velocity by conservation of momentum. This method of creating turbulence favourable to burning was originated by Harry Weslake in 1948. "Squish" resulted from local areas of close clearance at TDC in the chamber causing ejection of a turbulent charge flow towards the sparking plug. This had been patented by Harry Ricardo in 1919 to raise side-valve compression ratio and it had given 3 decades of new life to that cheap type, but he had not used it in his famous DOHC 1922 Vauxhall TT engine (see SO8 in Appendix 1). It is believed it was first used in an OHV engine by Leo Kuzmicki for the 1951 Norton 350 cc racing machine, with substantial benefit (683)(see **Sub-Note A**). It has been suggested that Gioachino Colombo adopted Squish in his improvements to the 1952 Maserati 2L engine; if so, Note 57 shows that this was *not* carried over to the 1954 250F 2.5L type, although ref. (32) has a sketch of such an arrangement (which clearly is *not* from parts seen).

After 1960 the then-top-quality of pump petrol at 102 Research Octane Number (RON) was specified by rule and this Octane rating has remained unchanged since, although with a steady tightening of the specification after mid-1992 to eliminate power-boosting constituents (see Appendix 2 and Note 90).

Keith Duckworth and Barrel Turbulence

Keith Duckworth in his 1965-designed Cosworth type FVA engine and later types, including of course his famous DFV, introduced an inlet port non-orthogonal to the valve head to promote *deliberately* "Barrel Turbulence" in the plane of crankshaft rotation for the same purpose as Axial Swirl but more effectively (see Note 80 and also "The Unique Cosworth Story"). [This in-cylinder motion is also called "Tumble Swirl" and sometimes the axial variety is simply called "swirl" and the crank plane version "tumble".] It is posible that some previous engines with side-draught ports had some tumble *accidentally.*

Duckworth used a narrower valve included angle (VIA = $40'^{0}$, reduced to 32^{0}) in his re-use of 4-valve-per-cylinder architecture to assist his barrel turbulence and to produce a compact combustion chamber with a flat piston top having squish plateaux and no hump to sub-divide the charge, plus the advantage of a central sparking plug.

Pressure-Charging and Toluene

With TurboCharging (TC) during 1983 – 1988 Toluene-base fuel was developed specifically to meet the *petrol* RON102 regulation test in low-speed laboratory engines but then give superior knock-resistance in high-speed racing engines (see Note 90). During this era R around 7½ was typical when maximum IVP and power were sought but rising to 9.4 in the 1988 Honda (Eg. 71) as IVP was restricted and race fuel quantity diminished (20).

Tighter fuel control from mid-1992

The chemical competition – which had made nonsense of the 1957 fuel companies cry of "petrol the same as you can buy" – continued into the post-1988 3rd Naturally-Aspirated Era (535) until, in 1992, Honda in the RA122E/B (SO20 in Appendix 1) were using R = 12.9 with a Bore of 88 mm and B/S = 1.84 (69). As mentioned, half-way through that season the rule-making authorities had had enough of it and a tighter control was instituted to try to return racing fuel to "real petrol". This cost the Honda 5% of power (69).

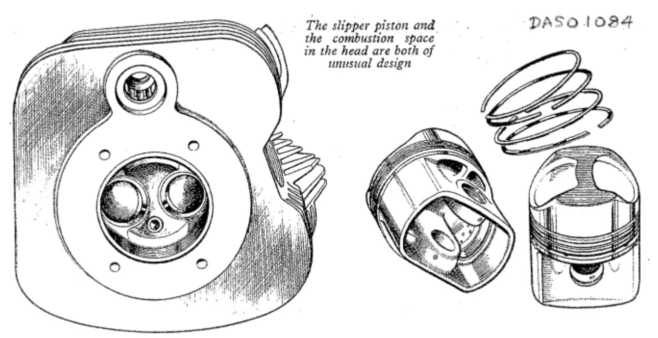
The last official figure for R for a GP engine in the review period was 12 in the 2000 Ferrari 049 (Eg. 85) with a bore of 96 mm and B/S = 2.32. Piston-crown cut-outs are of course required to permit all valves open at exhaust TDC. With tight-clearance squish plateaux this ratio also requires for reliability computerised stressing to calculate component elastic deflections and temperature differential expansions plus computer-controlled machining plus electronic engine-clutch-gearbox management (EMS) to avoid over-revving. The EMS and knock sensors are also needed to retard the ignition to avoid detonation at unfavourable throttle and speed combinations.

<u>Sub-Note A</u>

<u>"Squish" in the 1952 Norton 350 cc</u>

Geoff Duke's autobiography (683) describes how Leo Kuzmicki, a pre-WW2 lecturer on I.C. engines at Warsaw University was working post-WW2 as a labourer at Nortons. A mechanic told their racing engineer,Joe Craig, of this and Craig asked Kuzmicki to improve the works 350 cc after the 1950 season in which Velocette had beaten the Norton in that Championship class.

Kuzmicki altered the piston and combustion chamber of the 1-cylinder aircooled engine to provide squish. A drawing of the modified parts is given below.



This drawing of the 1954 version (1084) describes the parts as follows:-

"*...the piston has a completely flat crown..(it) must..be viewed in conjunction with the combustion space in the head..in the form of a shallow dome, the base diameter of which is less than the bore; hence there is a square step between the two. Because the piston crown comes up very close to the step there is a decided squish effect as top dead centre is approached*". [Note that this description was only made public in 1955, after Norton had given up full-time works racing.]

The gain from squish was mixed with a gain from compression ratio made possible by altering the head from a bronze "skull" with Al-alloy fins cast-on to an all-Al-alloy head, both engines on 80 ON petrol. If this was worth one unit of R, say from 7.5 to 8.5, the ASE benefit would have been 4%.The actual power increase was from 28 BHP @ 7,200 RPM to 36 @ 8,000, + 28.6% (683). [There was a small reduction in Stroke to keep the increase in MPSP down to 3.7% :-

1 a/c 73.336 x 82.5 mm altered to 75.9 x 77 mm.]

Squish therefore added around 25% of power.

The overall racing result was a 4.7% increase in Isle of Man lap speed (which is close to the [Power/ All-up Weight]$^{1/5}$ correlation established by the author for that circuit between 250 cc, 350 cc and 500 cc motorcycles in 1951, which would have shown + 5.2% for +28.6% power).

Note 24

Grand Prix 2-strokes

The only serious attempt to build a car GP 2-stroke engine was made by FIAT in 1925 with their type 451 IL6 1.5 L twin-crankshaft opposed-piston pressure-charged design aimed at the 1926 formula. This developed well over 100 HP/L on the test bed but overheating problems (burnt exhaust pistons and premature ignition of the charge) could not be overcome (66). See Fig. N24A.

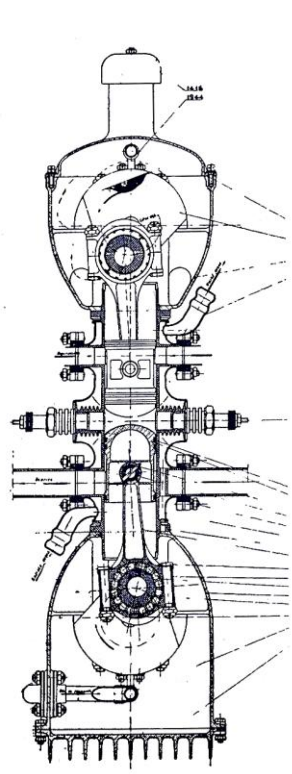
A pointed comparison between 4-stroke and 2-stroke engines was made by Joe Craig, racing engineer of Norton, for 1938 motorcycles racing in the Isle of Man TTs:- his 500 cc naturally-aspirated 4-stroke had won the Senior TT of 264 miles at 89.1 MPH and 30 miles per gallon (MPG); whereas the supercharged 250 cc 2-stroke DKW won the Lightweight TT over the same distance at 78.5 MPH and 20 MPG (12B)!

It can be observed, however, that in Grand Prix motorcycle racing some 4 decades later the 2-stroke supplanted completely the 4-stroke in all NA capacity classes by 1975. This was due largely to extra porting plus water-cooling and a highly-tuned exhaust system which were pioneered by Walter Kaaden of the East German firm of MZ in the late ‘50s and early ‘60s. By 1995 continuing development, (eg. of the Aprilia RSV250) had achieved the long-sought aim of a BMEP equal to that of the best 4-stroke -14 Bar – at similar MPS – 23 m/s – (762).While this development had overcome the quadruple 2-stroke problems of:-“breathing through its bottom-end”; limited inlet opening duration; mixing of inlet charge with exhaust residuals; and cooling difficulties with a power stroke each revolution, it was by using sharply-tuned resonant inlet and especially exhaust systems which so limited the useful range of RPM that the result would have been hopeless in a car. The exhaust emissions from a large 2-stroke would have been unacceptable in any case.

In 2001 the premier class of motorcycle racing (re-named “MotoGP”) allowed 990 cc 4-strokes to compete with 500 cc 2-strokes and at that capacity ratio of 1.98 the former quickly re-established its supremacy.

An article on "Grand Prix Motorcycle Engine Development, 1949 - 2008" which goes into the 4-stroke versus 2-stroke battle in more detail now available on this website.

Fig. N24A
1925 FIAT T451
2-Stroke
Opposed IL6 = 12 cylinders 52/58.5 = 0.889 1,491
DASO 66



Note 24B

Other Configurations

Doubling-up engines by gearing their crankshafts together inside a common crankcase was tried by FIAT with their type 406 in 1927. This had 2 x IL6 750 cc vertical blocks to give 1.5L, MSC. It was a 3 overhead camshaft engine, the inner inclined rows of valves being operated by a single central shaft (see Fig. N24B-A). It showed promise by winning a short Monza race but the firm then withdrew from motor-racing to concentrate on the Schneider Trophy.
The same 2-crank configuration was built by Alfa Romeo in 1938 as the type 316, using 2 x IL8 1.5L type 158 MSC blocks at 60^0 to give 3L. The cars secured 2^{nd} and 4^{th} places in the Italian GP.
The 1938 French SEFAC project had 2 vertical 1.5L blocks = 3L but it only appeared unsuccessfully a few times.

Twin engines in a chassis have also raced in *Formule Libre* events. Over 1929 to 1932 Maserati types V4 and V5 had 2 x IL8 totalling 4L and 5L. In 1931 the Alfa Romeo type A had 2 x IL6 1,750 cc engines to give 3.5L. These cars had side-by-side engines. A *tandem* arrangement was the Alfa Romeo "*Bimotore*" of 1935 with 2 x IL8 2.9L = 5.8L and later 6.3 L. The engine technology in these cases was within that described in the main text.

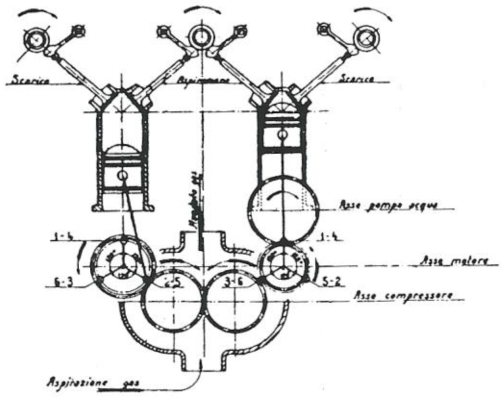
Probably the most surprising configuration ever conceived for Grand Prix racing was the 1955 Ferrari type 116 IL2 2.5L, designed by Aurelio Lampredi, which produced 70 HP/L on test (8) but was never raced because of its terrible vibration (see Fig. N24B-B).

Transversely-mid-mounted engines have been used on two occasions:-

- The 1956 Bugatti type 251 IL8 2.5L designed by Gioacchino Colombo, which raced once;
- The 1964 – 1965 Honda types RA271/272 60V12 1.5L which won the last race of the 1.5L formula.

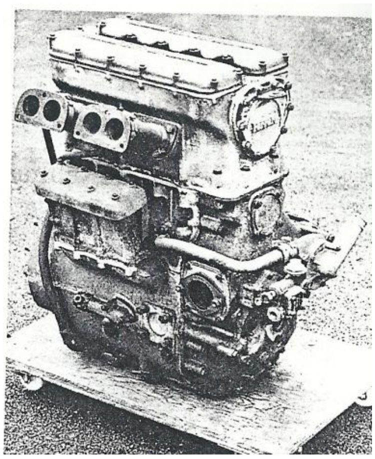
Maserati also built a 60V12 1.5L intended for transverse mounting in 1963, type 8F1, but it was never installed in a chassis.

Fig. N24B-A



1927 FIAT T406
U12 (2 x IL6 geared) 50/63 = 0.794 1,484 cc
MSC 187 HP @ 8,500 RPM
DASO 66 & 938

Fig. N24B-B



Classic & Sports Car June 1997

1955 Ferrari 116
IL2 118/114 = 1.035 2,493 cc
NA 174 HP @ 4,800 RPM
4 v/c

Note 24B

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MSC 187 HP @ 8,500 RPM
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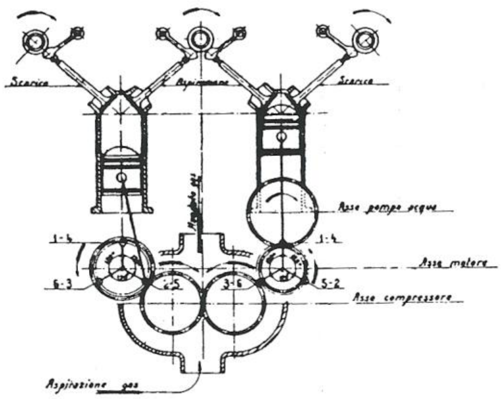
Fig. N24B-A



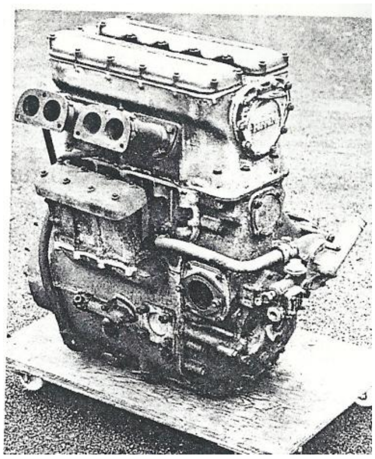


Fig. N24B-B

1955 Ferrari 116
IL2 118/114 = 1.035 2,493 cc
NA 174 HP @ 4,800 RPM
4 v/c

Classic & Sports Car June 1997

Note 25

Racing side-valve engines, 1906 – 1914

Renault pioneered the racing Side-Valve (SV) engine in their successful 1902 light car (2, 702).Only once has an SV engine powered a "Grand Prix Car-of-the-Year" (CoY) in the 1906 Renault type AK (see Eg.1 in the 1st Naturally-Aspirated Era). However, the same type of Renault might very well have won the limited fuel consumption 1907 French Grand Prix (FGP)((4), see Eg. 2) so it was not then outclassed completely in comparison with contemporary Overhead-Valve (OHV) designs. This was despite the basic disadvantages that:-

- The inlet and exhaust paths were even more tortuous (an extra 180^0 each of flow turning);
- Valves quarter-shrouded by the combustion chamber wall;
- Combustion chamber with a higher (Surface Area/Volume) ratio with higher heat losses.

Perhaps these factors were not critical while all multi-cylinder engines were then breathing through a single updraught carburetter. There *may* even have been some Combustion Efficiency gain in the SV by the greater mixing of fuel and air via the increased inlet turbulence.

The SV engines of the 1906 – 1914 period had claimed HP/Litre equal to, or greater than, the OHV competition. There must be some reason why they did not have greater GP success despite the simplicity of the type. Certainly a notable result was 3rd, 4th and 5th places in the famous 1912 *Formule Libre* FGP by Sunbeam IL4 3 L SV cars These had been entered primarily in the concurrent Coupe de l'Auto (Cdl'A) for light cars in which they were 1st,2nd and 3rd. They met and defeated decisively the 3 L Cdl'A version of the new 7.6 L DOHC Peugeot which won the GP.

This excellent result encouraged Louis Coatalen, chief of Sunbeam, to enter SV cars in the following year's FGP. The 1913 GP Sunbeam IL6 4.5 L engine was listed at 110 HP @ 3,000 RPM (24), where the winning Peugeot IL4 5.6 L had 115 HP, but the best Sunbeam finished 3rd, 2.6% slower. This was another limited-fuel race and the 1st and 2nd DOHC Peugeots averaged 14% inside the ration so that they had been driven well in hand. The SV Sunbeam was therefore beaten easily. Peugeot also had their revenge with 1st and 2nd in the 3 L Cdl'A race two months later, with improved 3 L DOHC cars, by beating the somewhat-revised SV Sunbeam into 3rd place.

Again, the 1914 FGP-winning IL4 4.5 L SOHC Mercedes is known to have had 104 HP (468) while Laurence Pomeroy was selling for the road Vauxhall 30/98s, IL4 4.53 L, with 95 HP (734). That power was at least confirmed by a post-WW2 test of a rebuilt 30/98 SV engine which showed 100 HP @ 2,800 RPM and 120 HP @ 3,500 (735) (some advantage in that test must have accrued from modern higher-Octane petrol permitting more ignition advance and also oil of lower viscosity).

Why, then, did LP Senior not enter tuned-up 30/98s in the 1914 FGP (reducing swept volume a trifle) but instead go to all the expense of a new DOHC design, alleged to give 130 HP (224) (but (371), more nearly contemporary, quotes 110 HP @ 3,000 RPM – a classic example of the pitfalls awaiting racing engine analysis!)?

A further definite case of SV achievement is that the 1st car in the world to better 100 miles in the hour, actually 103.8 miles (167 km) in February 1913 at Brooklands was the Talbot IL4 4.53 L single-seater with that valve gear, a hotted-up touring engine, driven by Percy Lambert.

Sustainable power

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*Brooklands Outer Circuit lap speeds rose with the cube root of Power/Weight ratio, for given body types, up to 135 MPH (732).

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Sub-Note A

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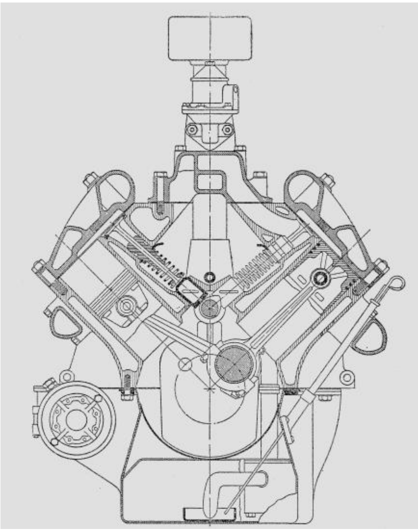
	NA, Petrol, at STP	Source
Date	1938	
Make	Ford	
Type	90V8 221 cid	
MDR	1	
V cc	3,621	
NP RPM	3,750	365
R	6.3	365
ASE	0.52	
EV	0.71	594
EC	0.59	From SFC = 0.62 lb/BHP. Hour @ NP (365), known ASE and given EM (594). BThE = 21.4%.
EM	0.7	594
PP BHP	88*	365

*As described in the main text this was a short-time rating.
The "continuous" rating was 45 BHP (736).

Comparison with Renault AK

The figures make a contrast with the one-and-only SV CoY, the 1906 Renault AK. The Ford was about the same Peak Power but obtained this from just over ¼ of the Swept Volume at 3 x RPM and with about 2 units higher Compression Ratio with the Al-alloy squish head on ordinary 1938 commercial Petrol – say 70 Octane rating instead of a (retrospectively-rated) 45 in 1906.

Fig.N25A
Ford 90V8 221 cid



Note 25

Racing side-valve engines, 1906 – 1914

Renault pioneered the racing Side-Valve (SV) engine in their successful 1902 light car (2, 702).Only once has an SV engine powered a "Grand Prix Car-of-the-Year" (CoY) in the 1906 Renault type AK (see Eg.1 in the 1st Naturally-Aspirated Era). However, the same type of Renault might very well have won the limited fuel consumption 1907 French Grand Prix (FGP)((4), see Eg. 2) so it was not then outclassed completely in comparison with contemporary Overhead-Valve (OHV) designs. This was despite the basic disadvantages that:-

- The inlet and exhaust paths were even more tortuous (an extra 180^0 each of flow turning);
- Valves quarter-shrouded by the combustion chamber wall;
- Combustion chamber with a higher (Surface Area/Volume) ratio with higher heat losses.

Perhaps these factors were not critical while all multi-cylinder engines were then breathing through a single updraught carburetter. There *may* even have been some Combustion Efficiency gain in the SV by the greater mixing of fuel and air via the increased inlet turbulence.

The SV engines of the 1906 – 1914 period had claimed HP/Litre equal to, or greater than, the OHV competition. There must be some reason why they did not have greater GP success despite the simplicity of the type. Certainly a notable result was 3rd, 4th and 5th places in the famous 1912 *Formule Libre* FGP by Sunbeam IL4 3 L SV cars These had been entered primarily in the concurrent Coupe de l'Auto (Cdl'A) for light cars in which they were 1st,2nd and 3rd. They met and defeated decisively the 3 L Cdl'A version of the new 7.6 L DOHC Peugeot which won the GP.

This excellent result encouraged Louis Coatalen, chief of Sunbeam, to enter SV cars in the following year's FGP. The 1913 GP Sunbeam IL6 4.5 L engine was listed at 110 HP @ 3,000 RPM (24), where the winning Peugeot IL4 5.6 L had 115 HP, but the best Sunbeam finished 3rd, 2.6% slower. This was another limited-fuel race and the 1st and 2nd DOHC Peugeots averaged 14% inside the ration so that they had been driven well in hand. The SV Sunbeam was therefore beaten easily. Peugeot also had their revenge with 1st and 2nd in the 3 L Cdl'A race two months later, with improved 3 L DOHC cars, by beating the somewhat-revised SV Sunbeam into 3rd place.

Again, the 1914 FGP-winning IL4 4.5 L SOHC Mercedes is known to have had 104 HP (468) while Laurence Pomeroy was selling for the road Vauxhall 30/98s, IL4 4.53 L, with 95 HP (734). That power was at least confirmed by a post-WW2 test of a rebuilt 30/98 SV engine which showed 100 HP @ 2,800 RPM and 120 HP @ 3,500 (735) (some advantage in that test must have accrued from modern higher-Octane petrol permitting more ignition advance and also oil of lower viscosity).

Why, then, did LP Senior not enter tuned-up 30/98s in the 1914 FGP (reducing swept volume a trifle) but instead go to all the expense of a new DOHC design, alleged to give 130 HP (224) (but (371), more nearly contemporary, quotes 110 HP @ 3,000 RPM – a classic example of the pitfalls awaiting racing engine analysis!)?

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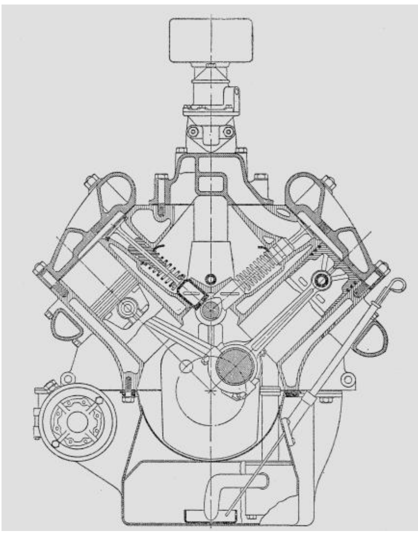

Note 25B

Inverted-cup tappets

The 14.1 L FIAT which finished 1.5% behind the 7.6 L Peugeot in the 1912 Grand Prix de l'ACF was a 1911-designed type S74 which had been victorious in the American GP of that year and which was developed from the similar 1910 type S61 of 10.1 L which won the Sarthe Club's Formule Libre race of 1911. In ref. (2) it is stated that Laurence Pomeroy Jnr.'s research showed that the S61 was an Ettore Bugatti design of 1909 "either directly or indirectly". The engine had 4 vertical overhead valves per cylinder operated by a single overhead camshaft via 2-valve bridge pieces, with inlets and exhausts alternating along the head. Cam side-thrust was taken by "inverted-cup tappets" fitting over the valve stems and the top of the springs, the 1st use of such a feature (see the picture Appendix to Note 15). Such tappets were patented by Albert Morin in 1916 (26) and were adopted by Ernest Henri with his double overhead camshafts in the 1919 Ballot for Indianapolis. They eventually became a popular "Car-of-the Year" method of connection between cam and valve stem in the late '50s to late '90s.

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Tumble Swirl: side ports and vertical ports

The inflow on suction through a poppet valve is biased by its momentum to the further side of the valve as it passes round the unavoidable curve to the valve head. With a 'side-draught' port (ie in relation to the cylinder axis) and with an inclined valve, part of the flow then impinges on the far wall of the cylinder and is caused thereby to loop or 'tumble' downwards. As the piston rises again on the compression stroke, the conservation of angular momentum while the rotary motion in the crankshaft plane is reduced in radius gives much increased circulation velocity to the mixture. This raises the rate of burning after ignition and so improves Combustion Efficiency (EC).

Swirl imparted around the cylinder axis by inlet ports curved in that plane, which was patented by Harry Weslake in 1948 when applied first to the Jaguar XK120 (214) (See sub-Note A), is less magnified by the rising piston compressing the charge into the combustion chamber.

'Tumble Swirl' (TS) in the crank plane is likely to have occurred naturally in many engines with inclined valves and side ports, but not by design intention. In the Cosworth FVA, designed by Keith Duckworth in 1965, TS was encouraged deliberately by having the outer part of the inlet passage non-orthogonal to the valve head by 20° and by increasing Inlet Valve Maximum Lift/Head Diameter to 0.3 from the then-usual 0.25. Volumetric Efficiency (EV) was sacrificed to some extent so as to gain EC and maximise (EV x EC). By having 4 valves-per-cylinder to provide the necessary areas the valve stems could be at only 20° to the cylinder axis, so that the inclination of the outer inlet passage wall at approach to the valve head was 20° + 20° = 40°. Therefore, the biased flow did not go straight across into the exhaust during the timing overlap period used to create extra suction. The central sparking plug made possible by four valves was placed perfectly to ignite the change whirling beneath it.

Sketches illustrating Axial Swirl and Tumble Swirl are given on P.2.

In the '30s, the two valves-per-cylinder 'flat port' motor-cycle head design also had non- orthogonal inlet passages near the valve head but, with a valve stem at 45° so that the outer wall angle was around 65°, they suffered from the charge loss to exhaust mentioned above, as pointed out by Phil Irving (76). The 1949 works 500cc Norton (which is believed to have been the same as the 1938 engine) had 27° non-orthogonality from a valve head with a stem at 37° to the cylinder axis, so that the outer flow angle was 64° to that axis (68). The later works Nortons (post 1950) had a 20° non-orthogonal wall but a valve stem at 32°, ie 52° wall inclination (480).

In the Cosworth DFV, designed one year after the FVA, the stem angle was reduced to 16° so that the flow struck the cylinder wall at a nominal 36°.

With an 'axial' (or 'vertical' or 'fully downdraught') inlet port, ie one at 0° to the cylinder axis, the inflow is biased by the inclined valve to pass directly into the cylinder, which raises EV but does not create TS and so (EV x EC) is lower than the 'side draught' engine. Walter Hassan, as a result of his experiments (going back to a pre-XKl20 design (214», was convinced that the axial port was not as good as the side port for producing maximum power, although it could be better at lower speeds (515). Harry Weslake also disliked the vertical port and the late Brian Lovell stated that this was because of its reduced contribution to rotational swirl in the cylinder (587). It is possible, however, that Weslake was considering that such a port could not be arranged tangentially to provide circumferential swirl.

Sub-Note A: Early use of axial swirl

Axial swirl was used before 1914 by Dr K Hesselman of the Swedish Atlas Co in Diesel engines for submarines in order to help fuel injection without air blast, which was the previous method.
This swirl was created by partial masking of the inlet valve circumference, however, not by the shaped inlet port later introduced by Weslake (852, 947).

After discussion with Ricardo in 1914, when the latter described his theory that petrol engine knocking arose in the end gas, Hesselman built engines with axial swirl in which the petrol (or even kerosene) was injected just upstream of the sparking plug so that the end charge had no fuel and could not detonate. These engines were successful and were also built in the USA (852).

Harry Weslake's Axial Flow

1948 Jaguar XK120

IL6 83/106 = 0.783 3.441 cc

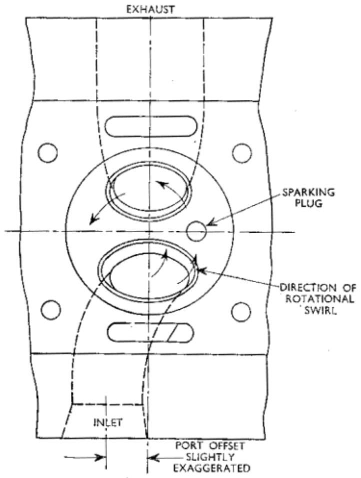
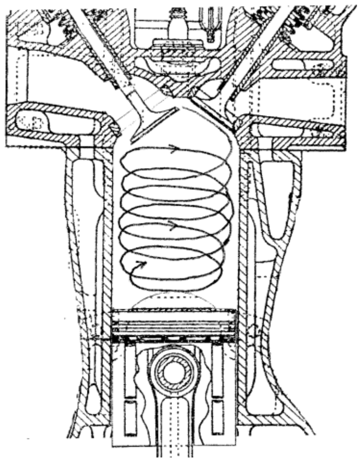


Fig. 9. Plan of Valve Ports

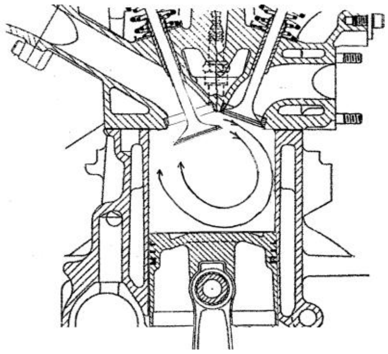


DASO 214

Keith Duckworth's Barrel Turbulence (aka Tumble Swirl)

1965 Cosworth FVA

IL4 3.375''/2.722'' (85.725mm/69.139) = 1.24 1,596 cc



Note 27

Tuning of individual inlet and exhaust systems

Naturally, single-cylinder engine makers were those who developed most intensively the art and science of tuning individual inlet and exhaust tracts for frequency and inertia effects in the air/mixture/gas flows so as to create a higher-than-normal density in a Naturally Aspirated (NA) piston engine cylinder at the moment the inlet valve closed, so raising Volumetric Efficiency (EV) and therefore BMEP. This was not actually 'free supercharge' because the gain at the tuned RPM had to be 'paid for' in a loss of EV at some other speeds - until variable length or variable capacity systems were introduced when the 'payment' was in terms of extra complexity and weight.

Pre-WW2 the British motor-cycle racing singles, especially Norton and Velocette, were the beneficiaries of the art, carried to the 'super tuning' level with diffusers ('megaphones ') on the exhaust to lower the pressure at the valve and help induce more charge. The 1938 Senior TT -winning Norton had an area ratio of about 8 from exhaust port to megaphone exit and the latter had an included cone angle of 13°. In this way Joe Craig, Norton's engineering/racing manager, obtained BMPP = 13 Bar at R = 11 on 50/50 petrol/Benzole fuel at MPSP = 21.2 m/s ((12B) and SO12). For comparison, the 1938 Delahaye GP Type 145 60V12 4.5L had BMPP = 9.4 Bar at R = 8.5, probably on similar fuel with three two-choke carburettors, at MPSP = 14.1 m/s (485). However, the Norton would not run smoothly below 60% of peak speed and the clutch had to be slipped at the start and on hairpin corners.

Continental motor-cycle racers in the mid '30s had gone instead of super-tuning for the allowed option of mechanical supercharging but Moto-Guzzi and Benelli had developed the NA tuning method.

Car engine tuning - the 1922 Miller 183

In car engines, inlet tuning had been pioneered by Harry Miller who, in 1922, had fitted his 183 cu. inch (3L) IL8 with individual tracts breathing through four two-choke own-design up draught carburettors. However, this engine was very limited in RPM, short of power peak, by having only a three-bearing crank. Its BMEP = 9.4 Bar on R = 8, probably petrol, at MPS = 13.6 m/s ((6) and S07))*. It is speculated by Griffith Borgeson that Eduardo Weber was inspired to produce his later famous two-choke carburettors by seeing the Miller units (on a 2L car) at Monza in 1923 (6).

As Miller, in company with all other front-rank car racing engine makers, moved on to mechanical supercharging in 1924, his NA work came to a halt. Later NA engines from the Offenhauser/Meyer-Drake successors to Miller did not resume individual and tuned inlets until the development of Hilborn indirect fuel injection in 1950.

Car engine tuning - the Dixon Rileys 1930-1935

Freddy Dixon, a very experienced and successful motor-cycle racer, following up a works Riley initiative in fitting four Amal motor-cycle carburettors to an IL4 1100cc sports car engine in 1930, became the great exponent in the mid '30s of individual tuned inlet and exhaust systems on NA Rileys from the 1l00cc to the lL6 2L. With the latter in a low nominal- two-seater, long-tail chassis he achieved a lap of the Brooklands Outer Circuit (BOC) banked track at 134.4 mph (216.3 kph) in 1935. This may be compared with the all-time BOC record of 143.4 mph made by a 500 HP machine and is believed to represent a genuine 150 HP from 2L NA (BMPP = 11.3 Bar at R = 10.8 on alcohol/toluol fuel, MPSP about 21 m/s) (141) (Dixon never bench-tested his engines). Percy Maclure and Hector Dobbs built similar Riley-based engines.

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*See the note at the end concerning the tuning of the Miller 183.

1950 developments, British and Italian

Ferrari, racing V12 F2 2L NA engines in 1950, was impressed by the performance on twisty circuits of the IL4 cars of Amadee Gordini and John Heath, the latter named HWM and fitted with Geoffrey Taylor-built Alta engines. Both raced with only two carburettors. The Alta was tested originally with four Amal carburettors and four individual, tuned-length exhaust pipes and as such carried on and improved upon the pre- WW2 Dixon layout - but the Amals did not give adequate starting or acceleration characteristics and were not raced, while breakages of the long separate exhausts soon forced a change to a manifold and single pipe; power dropped 13% after these two alterations (147).

Connaught at the end of 1950 had introduced an F2 car with four tuned inlets using Amals applied to an IL4 1.8L modified Lea-Francis engine (originally aimed at US Midget racing) very similar in head design to Dixon's Rileys. The exhaust system had cylinders 1,4 and 2,3 flowing into a separate pair of tail pipes. This car had not raced abroad.

When Aurelio Lampredi designed the Ferrari Type 500 IL4 F2 2L NA engine in late 1950, as his response to the Gordini and HWM performances, it is considered that, because of the time, space and culture gaps from the earlier work on tuned, individual inlet and exhaust systems, he made his own decisions in the matter of such tuning. Weber apparently produced a suitable two-choke/one-float-chamber horizontal carburettor (Type DCO) especially for the new Ferrari engine at the appropriate bore (50mm) and port spacing.

With a bore of 38mm the DCO was sold simultaneously to HWM for their 1951 single-seaters. On their 2L Alta engine these Weber units became part of a tuned individual inlet and stub-exhaust system which was actually the first to race internationally post-WW2, 29 years after Miller's pioneering engine (it recovered the 13% lost from the early 1950 tests, back to 130 HP) (147). The Ferrari IL4 engine, in 2.5L form, first raced in early September 1951.

Simple inlet-tuning theory

In the simple organ-pipe theory, as given in (282), the Mean Piston Speed (MPS) at which resonance occurs can be estimated from.-

$$MPS = 88.25 \times (S/LIN) \text{ m/s}$$

where LIN = Inlet system length to back of valve, in same units as S = Stroke.

Consequently, the following applies:

LIN/S	3	3.53	4	5	6	7
Resonant MPS m/s	29.4	25	22.1	17.7	14.7	12.6

The Ferrari Type 500 in 1953 (Eg 31) had LIN/S = 330/78 = 4.23 and would have resonated theoretically at 20.9 m/s; the peak speed was actually 19.5 m/s.

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* Miller 183

Ref (6) shows that this engine had LIN/S = 600/101.6 = 5.9 and the simple formula gives resonant MPS = about 15 m/s, versus the rated maximum of 13.6 m/s. The optimum length was found by varying the pipe lengths and timing the car round the Beverly Hills banked board track.

Note 28

Fabricated-steel engine construction

The 1st internal combustion engine with fabricated-steel stationary structure was probably that built by Charles Manly in the USA for Langley's unsuccessful aircraft in 1902. This was a water-cooled radial 5-cylinder 5'' Bore x 5.5'' Stroke, 8.8 Litre design. It weighed only 151 lb (68.5 kg) and held 52 HP for 10 hours in 1904, having 0.76 HP/kg (726). This was achieved after Manly had visited many European firms in 1900 and had been advised they could not make a 12 HP engine under 100 kg, i.e. only 1/6th of the power/weight ratio which he obtained 4 years later. As an aside, it is noted that the Wright brothers, with Charles Taylor, likewise had to build their own aviation power-plant after discouragement by commercial makers. Their cast-Al-body, steel-linered IL4 of 4'' Bore x 4'' Stroke only sustained 12 HP at a weight of 170 lb (77 kg) (592, 722), 0.16 HP/kg or 1/5th of Manly's engine. Nevertheless, their aircraft was, of course, successful in December 1903. It is a pity that the two groups could not have pooled their knowledge since the Manly had a 40 HP output as early as April 1902. It was not the fault of the engine that the Langley "aerodrome" failed – it crashed twice in 1903 directly after unpiloted catapult launch because of structural weakness.

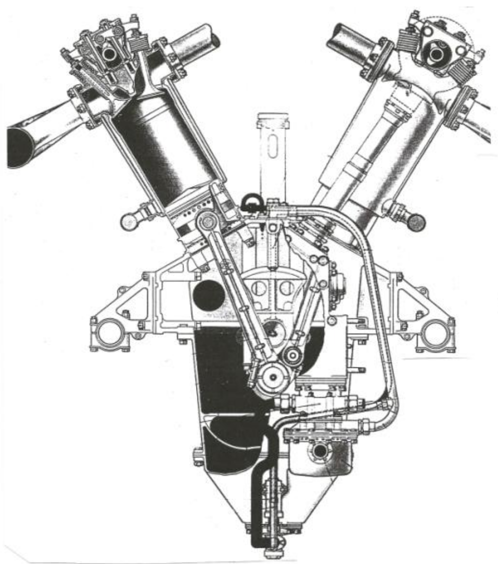
For comparison, the cast-iron 1902 Panhard racing engine, IL4 13.7 L, produced 70 HP at 0.22 HP/kg, with the non-scaling weight advantage of large size (2, 4).

All the above-mentioned engines had suction-operated inlet valves.

The Manly engine was fully described, with drawings, in a Smithsonian memoir of 1911 (726).The German "Kaiserpreis" aero-engine competition rules were issued in May 1912 and Daimler then entered their DF80 design with fabricated-steel upperworks (468). It seems entirely possible that Paul Daimler had studied the Manly design and added the improvement of welding instead of brazing. Of course, overhead mechanically-opened inlet valves had become normal by then.

The most important users of a constructional system similar to the Daimler's were Rolls-Royce (see Fig. N28A), FIAT and Liberty WW1 aero engines and FIAT racing engines of the early '20s – which were then copied by Sunbeam and Alfa Romeo.

Fig. N28A
1916 Rolls-Royce Eagle Series 1
60V12 4.5''/6.5'' = 0.692 1,240.5 cid
(114.3mm/165.1 20,329 cc)
243 BHP @1,600 RPM (later uprated to 255 BHP @ 1,800 RPM)
Designed by Henry Royce, assisted by Albert Elliott and Maurice Olley
DASO R-RHT Historical Series No. 43



Note 29

The 'Nikasil' process

The 'Nikasil' surface treatment process was developed in the late '60s by Mahle, the German piston manufacturer, in conjunction with NSU and Daimler-Benz (468) originally to provide a wear resistant coating for Al-alloy rotor casings fitted to Wankel rotary engines. It was the result of a long programme of material combinations tried by many Wankel licencees to overcome a 'ripple' wear pattern characteristic of this engine type.

The process used electrolytic deposition to plate the Al-alloy casings with nickel in which particles of silicon-carbide smaller than 1 micron (0.001mm) were dispersed. After finishing treatment the plating was only 200 microns (0.2mm) thick in the Wankel application (870).

Porsche first used 'Nikasil' coating in a normal piston engine for (air-cooled) Al-alloy cylinders in the 1971 5L development of the F12 Type 912 unit and, compared with the previous Cr-plated Al-alloy cylinders ('Chromal') found it increased power (302). The layer may have been thinner than that used in the Wankel engine (a 'few hundredths of a millimetre' was quoted in (241) when applied in 1973 to the Porsche Type 911/83 competition engine (also air-cooled).

The reported improvement in oil consumption of the DFV with 'Nikasil'-treated Al-alloy cylinders may have been a consequence of more rapid and complete bedding-in of the piston rings, which is what is thought to have occurred in the Porsche 912 engine. If this then reduced combustion gas blow-by a power gain would have been observed

Note 30

Increased Road Grip

Increased road grip has been generated since 1906 in 3 ways:-

(1.). Early road surfaces rapidly became loose on corners during races – the 1921 French Grand Prix was characterised by the winner, Jimmy Murphy, as "*a rock-throwing competition!*". From the mid-'20s asphalt and tar were used generally to bind the surfaces, although melting in excessive heat could still cause track problems as late as the 1959 French GP.

(2.). Tyre grip has improved steadily.

In 1954 Mercedes calculated cornering speeds on an assumption of a maximum friction coefficient, (Side load/Down load), of 1. Measurements justified this choice.

From 1961 high-hysteresis tread compounds were introduced to raise friction coefficient

Over 1964 – 1968 particularly there was a large increase in tyre (Width/Diameter) ratio to reduce slip angle at the limit of adhesion by providing a more effective contact patch shape, the (Lateral width/Circumferential length) ratio being raised. [This the tyre makers called "L*ower aspect Ratio*" by reference to (Radial Depth/Width) but which aerodynamicists by analogy with wing performance, Lift versus Incidence, would call "*Higher Aspect Ratio*"!]

Slick, i.e. pattern-less, tyres which produced a further reduction in contact patch distortion came into GP use for dry conditions in 1971. US Dragsters had used them, for acceleration benefit of course, since the early 1960s.

Radial-ply carcases, again to reduce contact patch distortion, began to supersede cross-ply from mid-1977 when Michelin pioneered them for racing (on the new Renault TurboCharged car) as they had done for road use some 15 years earlier. They were first used on a Championship-winning car by Ferrari in 1979.

Working in the *opposite* direction to improvement in road grip, rules were introduced in 1998 (which amplified earlier tyre width restrictions) to require *circumferential* grooves in treads to *promote* contact patch distortion and so cut cornering speeds. The details were 3 front tyre grooves and 4 rear. However, "*Tyre Wars*" between Goodyear and Bridgestone with competing tread compounds and carcase construction quickly negated the intended speed reduction. The grooving was changed in 1999 to 4 grooves all round.

Ref. (987) shows that in 2000 Bridgestone tyres for Ferrari were capable of a maximum friction coefficient of about 2.

(3.).Aerodynamics were harnessed deliberately to increase downforce for GP road-racing in 1968 by adding "*upside-down wings*". Ferrari led the way.

Lotus in 1977 pioneered the use of under-body venturis, made really effective by track-touching sliding skirts which prevented air inflow at the sides, to gain further downforce.

Both aerofoils since 1968 and venturis since 1977 have had many and various rules applied to limit their effect, especially the banning of skirts, but both are still present to contribute very largely to road grip.

The magnitude of the improvements in road grip over the review period can be shown in braking performance – perhaps only 1/4 g for rear-wheel-only drum brakes on loose surfaces in 1906 to 4 g from 350 kph for the Ferrari with Carbon-fibre-reinforced-carbon discs/ carbon pads at Monza in 2000 (987).

Note 31

Castor-base oil

In 1954 Mercedes-Benz found to their surprise that they had to revert from mineral to castor-base oil (i.e. vegetable-base) oil, supplied by Castrol, to cope with the surface loadings in their roller-and-ball-bearing M196 engines with desmodromic valve gear (468).

Note 32

1923 Sunbeam: Exhaust valve condition post- French Grand Prix

As in the main text, Callingham of Shell (294) stated that the winning Sunbeam had exhaust valves so badly burnt at the end of the race that it was doubtful if the car could have raced much further. Certainly it is agreed in Segrave's autobiography (763) that *Guinness'* engine had burnt valves, so that he stalled it at a hairpin on the last lap and had trouble re-starting, thereby losing a place and finishing 4th, but Segrave says he drove his own car which "*was in excellent condition*" back to England. It is the case that he had been unable to use above 90% of peak revs (756) for 86% of the race (763) without clutch slippage because of a faultily-fitted back stop but this then broke off and he had full power. The restriction must have saved his engine

The statements of Callingham and Segrave *are* reconcilable, bearing in mind that the winning engine would have had its block-cum-head removed after the race for officials to check the dimensions (Callingham, as a technical representative, may have been present). This would have offered the opportunity for the Sunbeam mechanics to do a top-end overhaul before Segrave returned to Wolverhampton.

On the other hand, Callingham, writing 13 years later, may have simply misremembered which engine had suffered.

Note 33

The banning of Continuously-Variable-Transmission (CVT)

With the final agreement in August 1993 of the changes to the technical rules for 1994, which *inter alia* banned automatic gear-changing, Williams were forced to abandon a CVT which they had planned for that season and had tested in an FW15C. Many years of research had been invested in it, in conjunction (it is believed) with the Dutch firm of Van Doorne Transmissie (VDT). This firm had been founded in 1972 after the original Van Doorne Automobiel Fabriek (DAF) which had pioneered a rubber-belt-linked double-opposed-expanding-cone approach to CVT for a small road car in the late'50s, had sold its car division to Volvo. DAF had actually raced an F3 car with the system in 1967 and won 2 races. The Van Doorne CVT later used a multi-segmented metal belt to replace the rubber part.

Had this Williams CVT been a success, i.e. the benefit overcoming the power-loss and reliability problems associated with previous step-less drive-ratio systems, there would have been an associated and significant effect on engine design since it could then have been optimised for maximum power at constant RPM. It has to be said that the steady engine noise would have been less thrilling!

Note 34

Optimum gas velocity at inlet

The reason why an optimum gas velocity occurs in the inlet flow of a liquid-fuelled four-stroke piston engine was explained by L Mantell in the '30s (294, p 82) as follows:-

> 'To *return ... to the inlet valve functioning as a disintegratory agent"* (*of the liquid fuel*) *"..., while making this big enough to pass a quantum of live charge, it must on the other hand be small enough to speed up the charge locally and so put the final finishing touch on the so-called atomisation of the fuel. This valve, therefore, has a critical size like the choke; <u>too big a valve may cause a loss of power just as an undersized valve will</u> (DST underlining) – in the former case the final state of the fuel in the head will be too coarse to effect a sufficiently complete evaporation and admixture with the oxygen. The result will be that all the large particles of (fuel) will burn too slowly to have their full driving effect ...* "

This descriptive approach is, in effect, pointing out that, in choosing inlet valve size, both 'breathing' and 'burning' have to be considered, ie what matters is maximising the product of Volumetric Efficiency (EV) x Combustion Efficiency (EC).

Inlet pressure loss rises as $(\text{velocity})^2$ and, regarding the effect on EC, C. F. and E. S. Taylor (594) published in 1938 tests which, by varying inlet port sizes at constant RPM, showed that flame speed rose linearly with inlet velocity. This illustrated the effect of in-cylinder turbulence as well as the better fuel air mixing referred to by Mantell.

It is apparent from the above that the optimum value of Mean Gas Velocity (MGV) at inlet will depend on:-

- The 'drag coefficient' of the inlet system (higher drag will lower the value). Up to 1952 all CoY engines had 'Tortuous' high-drag systems, and afterwards all had 'Individual, Tuned' systems of low drag (at resonant speed);
- The degree to which the fuel is 'atomised' by the method of supply, eg carburettor or injection (inlet or in-cylinder) or whether a supercharger is fitted which vaporises the fuel by its heat of compression and 'mashes' it mechanically in the rotor(s) (more 'atomisation' or evaporation pre-valve will also lower the value).
- The type of fuel (easy or hard to vaporise -the latter will raise the value).

MGV as a valuable guide to efficient engine design was probably first pointed out by Laurence Pomeroy Senior when Chief Engineer of Vauxhall around 1910. Ricardo has reported (343) that Pomeroy at that date *"contended that piston speed was limited only by breathing capacity and that (there was) ... no reason ... why piston speeds up to 2500 ft/min (12.7 m/s) should not be obtainable with a single row of valves and ... 3000 ft/min (15.2 m/s) or even over if a double row be employed.* "Pomeroy at that date was racing side valve engines, but would have known of Continental overhead valve (OHV) designs, especially, perhaps, the 1909 Fiat S61 engine with four vertical valves. If it is assumed that his comments applied to such OHV heads, with vertical valves contained within the cylinder bore, then it can be deduced that he was calculating from an optimum MGV around 200ft/sec (61 m/s) (based on valve <u>head</u> area, which has been standardised for this review because it is the figure most often available. Caution is needed on some early data where 'valve diameter' really means 'port diameter' and adjustment is needed. MGV is calculated from MPS with the standard assumption of incompressible flow). This MGV would have been for a 'Tortuous' inlet system, four cylinders drawing from one updraught carburettor.

Ricardo around 1918 took up Pomeroy's view that MGV was a critical factor in determining performance. In his later *High Speed Engine* (242) he published a generalised curve of EV v. MGV from which it can be deduced that Maximum Indicated Power per unit of inlet valve area, which is proportional to (EV x MGV) would occur at MGV = 250 ft/sec (76 m/s). Again, this would be supposing a 'Tortuous' inlet system and, with decreasing Mechanical Efficiency (EM) with rising RPM, the Maximum Brake Power would be at lower MGV. However, this result took no account of the effect of velocity on EC. Ricardo's 1922 IL4 3L racing design for Vauxhall, which bettered his general EV v MGV curve at the top end, and which, of course, integrated the velocity effects, peaked in BHP at MGV = 173 ft/sec (52.7 m/s). Although this had a dual-throat carburettor feeding cylinders 1,4 and 2,3, the approach to this was (a) very Tortuous and (b) warmed, by passage through the crankcase.

Mantell's deduction of the mid '30s has been quoted already. He gave no valve-related velocity but proposed an inlet peak value of 180 ft/sec (54.9 m/s) and a carburettor choke figure of 300 ft/sec (91.4 m/s).

Shortly after WW2 a statistical analysis of various aero engines was published in the USA (783), which gave the relation:-

$$A \qquad 0.025 \cdot \frac{(VN)}{(2000)} \text{ sq in}$$

where A — Inlet area based on the inner valve-seat diameter (neglecting the valve stem)

V — Swept Volume - cu in

N — RPM

This result, which applied to NA and PC units with B/S between 1.12 to 0.79, can be reduced to:-

MGV = 180ft/sec (55 m/s)

The inlet conditions would be 'Tortuous'.

Harry Mundy in 1957 (52) proposed an inlet port area ratio equivalent to IVA/PA = 0.33 for the Individual, Tuned case, ie at MPS = 4000 ft/min (20.3 m/s) which he then recommended as the mechanical limit, MGV = 202 ft/sec (61.6 m/s). At a piston speed typical of 1990 onwards, ie 5000 ft/min (25.4 m/s) MGV rises to 253 ft/sec (77 m/s).

C. F. Taylor in 1968 (784) modified the inlet velocity concept by introducing the valve coefficient of discharge to obtain the actual velocity from the nominal average and then related it to the speed of sound as a Mach Number (labelled Z). This, as for Ricardo and Mundy, did not introduce the effect of velocity on EC. The Z = 0.58 proposed for a racing engine with Individual, Tuned tracts and methanol fuel (cooling the charge and lowering the speed of sound) was equivalent to MGV = 70 m/s (Vol 2, p400). The compressible flow parameter (Mass Flow x -√(Temperature)/(Area x Pressure) at 58% of the speed of sound is 82% of the choking figure (429).

Finally, a practical case is known where the late Brian Lovell put to good use the general theory of an optimum inlet velocity. Newly arrived at Weslake's in the late '60s, he was asked to look at a 500cc twin which had given disappointing power. With specialist carburettor experience at Zenith, where the design guide was 'Optimum Mixing Velocity', he decided that the engine ports, sized to give 'Minimum Pressure Drop', were too large. Despite scepticism, they were reduced and a large power increase was obtained (782).

Statistics 1906-1998

The attached figures below on page 4, 111/DST and 112/DST, show I V A / P A and MGVP (MGV at Peak Power) for CoY engines over the years. This suggests that the best designers understood in the years of Tortuous inlet tracts, and usually with superchargers, that MGVP around 55 m/s was the value to aim at. When Individual, Tuned systems, normally-aspirated came into use, 1952 onwards, designers felt their way to about 75 m/s noting that inlet fuel injection came in 1962. The particularly low values for 1954-55 are those of the Mercedes M196, which had in-cylinder injection spraying partially onto the exhaust valve. With this process supplying all the necessary 'atomisation' of the methanol-base fuel, the inlet system could be sized to give 'minimum pressure drop' and raise EV without spoiling EC (at least not on account of insufficient flow speed through the valve – there were <u>other</u> spoilers in the M196, see the Design Era text, Egs 32 and 33).

The turbocharged engines of 1982-1988 did show lower values of MGVP, which is in accordance with the theory that the heat of compression, though largely removed by intercooling, provided some of the needful fuel/air mixing.

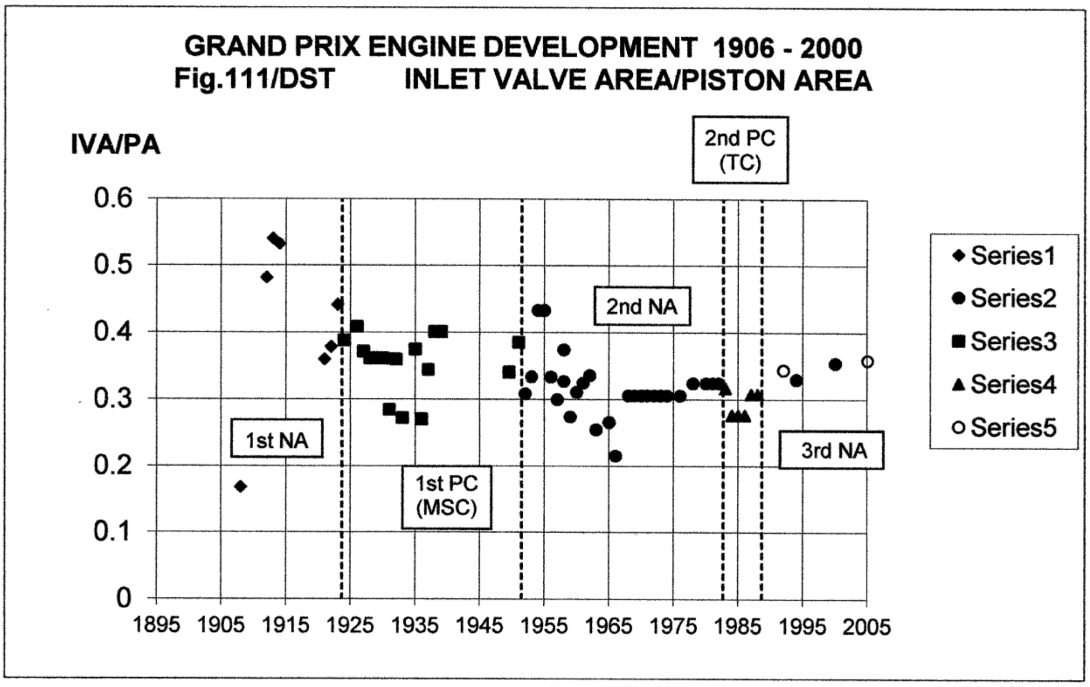
P.S.on Direct Petrol Injection (DPI)

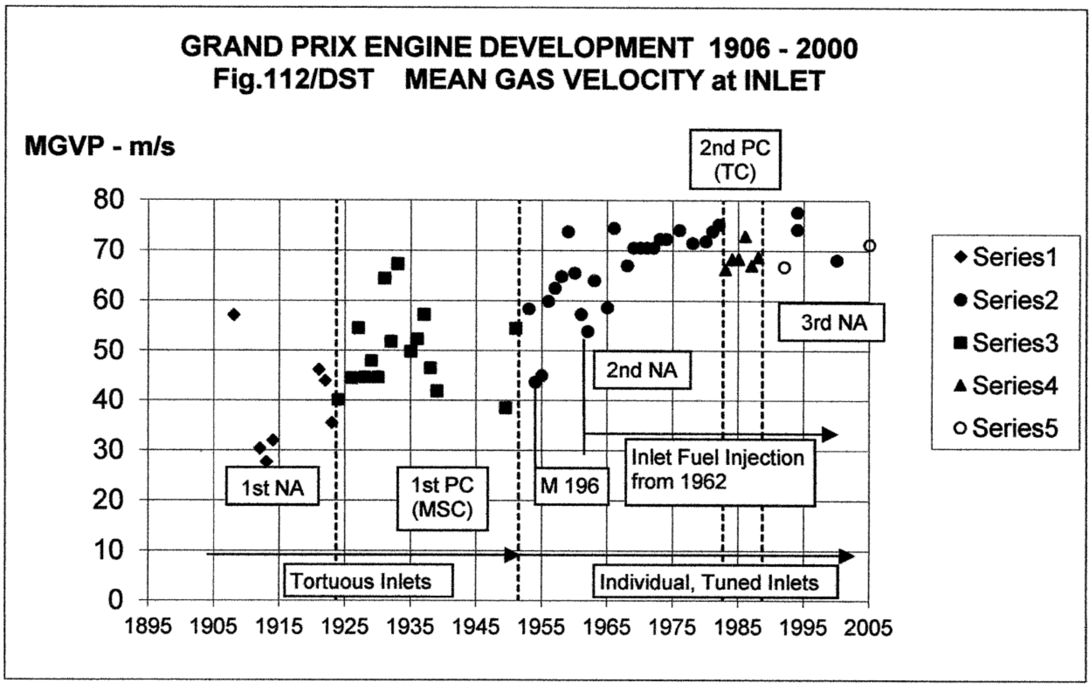
November 2013.

The point on P.1 about atomization of fuel is relevant to the 2014 rules which require Direct Petrol Injection (DPI) into the cylinder and allow a 500 Bar pressure for this system. This contrasts with only *7 Bar* in the 1967 Cosworth DFV Lucas *port* injection.

If this ultra-high-pressure DPI system can provide all the necessary preparation of the fuel/air mixture to obtain maximum Combustion Efficiency (EC) then there is no longer a need to compromise the inlet port and valve regarding velocity or the creation of swirl. They can be sized and shaped for maximum Volumetric Efficiency (EV).

Page 4 continues below


GRAND PRIX ENGINE DEVELOPMENT 1906 - 2000
Fig.111/DST INLET VALVE AREA/PISTON AREA
IVA/PA
2nd PC (TC)
0.6
0.5
0.4
0.3
0.2
0.1
0
2nd NA
1st NA
1st PC (MSC)
3rd NA
1895 1905 1915 1925 1935 1945 1955 1965 1975 1985 1995 2005
Series1
Series2
Series3
Series4
Series5


GRAND PRIX ENGINE DEVELOPMENT 1906 - 2000
Fig.112/DST MEAN GAS VELOCITY at INLET
MGVP - m/s
2nd PC (TC)
80
70
60
50
40
30
20
10
0
3rd NA
2nd NA
Inlet Fuel Injection from 1962
1st NA
1st PC (MSC)
M 196
Tortuous Inlets
Individual, Tuned Inlets
1895 1905 1915 1925 1935 1945 1955 1965 1975 1985 1995 2005
Series1
Series2
Series3
Series4
Series5

Note 35

The influence of Maurice Sizaire on piston engine design

The common 1906 belief that in piston engines a limit of 6 m/s (1,200 feet/minute) applied to the Mean Piston Speed [MPS = 2 x Stroke (S) x N RPM], because of inertial stresses (proportional to MPS^2) in the then-usual cast-iron pistons, was based on tests by the Automobile Club de France (ACF) of a wide variety of engines built up to that date (4). Changes in the piston material or its geometric proportions were not considered.

The consequence of a belief in a constant, limited, MPS can be seen by considering it in relation to the basic power equation:-

Power (P) proportional to [Brake Mean Effective Pressure (BMEP) x Swept Volume (V) x N]
where V = [Piston Area (PA) x S]
therefore P is proportional to [BMEP x PA x MPS].

In 1906 there was no discussion of how BMEP could be increased from existing levels, e.g. by raising Compression Ratio (R) above the range of 3 to 4 enforced to avoid detonation by the petrol of the day, because the influence on that parameter of fuel, inlet charge conditions and combustion chamber shape, etc, was unknown.

Therefore, taking BMEP for granted and then assuming a constant limiting value of MPS led the French organisers of the 1907 – 1908 Coupe de l'Auto Voiturette races to set only PA limits for power control. For a single-cylinder the maximum bore permitted was 100mm More generous PA was allowed for twins (+28%) and more again for fours (+69%), partly, perhaps, by perceptions of their respective lower efficiencies from higher area/volume ratios and more moving parts. More probably this generosity was *intended* to lead designers to smoother and more touring-useful multi-cylinder engines but, if so, it did not achieve its aim for several years.

Maurice Sizaire was the designer of the 1906 Coupe winner built by Sizaire et Naudin, which was 1 cylinder of Bore (B)/ S = 120mm/110 = 1.09 which had achieved an MPS of 7.3 m/s and Peak Power (PP)/ PA of 0.16 HP/cm^2. Not being a college-educated man – he had been a builder's draughtsman (361) – he had no respect for theoretical dogma on MPS. Instead he saw "Free Stroke" as an opportunity. He built in 1907 a 1 cylinder engine of 100mm/150 = 0.67 with *machined-steel pistons* which ran up to MPS = 12 m/s and produced 0.28 HP/cm^2, and this won the Coupe and also took 2nd place. In 1908 he went further still with a 1 cylinder engine 100mm/250 = 0.4 reaching 20 m/s and giving PP/PA = 0.53 HP/cm^2, again taking 1st and 2nd places in the Coupe (data from (259, 361); other details are given in Sub-Note A).

There *was* piston trouble in 1907. Maurice Sizaire admitted in 1964 (361) – 57 years after the event! – that the pistons had been machined too thinly and had to be changed during the race. This was done by driving each car into a wood alongside the long circuit, where it was an easy matter to fit new pistons to single cylinders having only 4 holding-down studs. Clearly, spares had been smuggled onto the cars in anticipation of a change, although it was against the rules. The 10 minutes time loss was explained to the stewards as drivers' calls of nature! However, this piston problem was *not* fundamental, as proved by the 1908 success at even higher MPS, noting that this race was 400 km long. Of course, the maximum MPS would not be used continually. The 1908 S & N level of 20 m/s (4,000 ft/min) *did* remain roughly the new limit for MPS for 6 decades afterwards, depending on race length and on Al-alloy pistons post-WW1 and whether Naturally-Aspirated or Pressure-Charged.

These 3 Sizaire engine types apparently were limited in RPM by their valve gear, since they ran at a constant Bore Speed [B x N] of 4 m/s. Therefore, [(PP/PA) x(B/S)] was fairly constant after allowing for R having been raised from 4.2 to 5 by 1908, worth nearly 9% of BMEP (see Sub-Note B). The reasoning on Bore Speed as a surrogate for Mean Valve Speed is given in Note 13 Part II.

The proof from these Sizaire engines that MPS was not then a limit under 20 m/s was quickly grasped by other Voiturette designers, including Michaux of Peugeot who even used B/S =0.25 in 1910.

The AC de la Sarthe clearly had these top-heavy Voiturettes in mind when setting the rules for a class in their projected 1911 race which restricted 4 cylinder B/S to 110mm/200 = 0.55 (V = 7,603 cc).

The new Peugeot racing car designer Ernest Henri began work to those rules but industry politics caused Peugeot to withdraw in 1911. However, the resultant Peugeot L76 EX1 then raced successfully in the 1912 Grand Prix de l'ACF and (with many other innovations) set a new standard for design, using MPS = nearly 15 m/s for that 1540 km race. The last previous Grand Prix winner in 1908, a Mercedes, had an MPS of just under 10 m/s for 770 km.

Frederick Lanchester

Frederick Lanchester in 1906 also had the same belief that RPM were limited by piston inertial stress and used Dimensional Analysis and piston geometry typical of the period (assuming that piston height would remain proportional to Bore) to suggest a "Rating Rule" in which the basic PP/PA was to be multiplied by **√**[B/S] to produce, he expected, a constant number for all engines (he offered later an alternative multiplier of **√**[(B/S)+ 0.3] (369).

Unfortunately for Lanchester's "Rating Rule" the Sizaire results disproved his theory shortly after it was published (see Sub-Note B). It is fair to say that he had not envisaged the grotesque engines which were developed for Voiturette racing.

The RAC Rating Rule

Also in 1906, the Royal Automobile Club (RAC) adopted a production engine piston area Rating Rule, slightly altered from a proposal by Prof. Dugald Clerk, which was equivalent to a PP/PA of just over ½ HP per square inch (0.079 HP/cm^2). HM Government accepted this RAC Formula, which was

$$PP = [0.4 \times \text{No. of cylinders} \times (B \text{ inches})^2]$$

for taxation purposes (on the principle that "people should pay for power"!) and held to it doggedly until 1947 in the teeth of evidence that the original averages used in the formula of MPS = 1,000 feet/minute (5.08 m/s) and BMEP = 67.2 psi (4.64 Bar) were being steadily left behind during 40 years of development of UK series-production engines. A pre-WW2-designed "bread & butter" example of 1947, the Morris "Ten" RAC HP having B/S = 63.5mm (2.5inches)/ 90 = 0.7 (V = 1,140 cc) ran at 2,700 ft/min (13.8 m/s)(+170%) with a BMEP of 92 psi (6.35 Bar)(+37%) to produce 37 HP @ 4,600 RPM.

In 1947, with the intention of inducing manufacturers to build engines more suitable for export markets, HMG then switched to a flat rate tax, irrespective of power. The first engine designed taking note of the new tax rule was the 1947 Standard "Vanguard" with B/S = 85mm/92 = 0.92 (V = 2,088 cc). This ran at 2,400 ft/min (12.3 m/s) and BMEP of 106 psi (7.29 Bar) to give 68 HP @ 4,000 RPM.

In 1947 an MPS of 2,500 ft/min (13 m/s) was considered a sound figure for a long-life production engine, where 4,000 ft/min (20 m/s) would be acceptable for a short-life Naturally-Aspirated pure-racing unit, both types using Al-alloy pistons.

Sub-Note A

Sizaire et Naudin Voiturette engines

Data sources (259,361).

Date	No. Cyl.	Bore (B) / Stroke (S) mm	Swept Volume (V) cc	Compression Ratio (R)	Peak Power (PP)* HP	@ RPM
1906	1	120 / 110 =1.09	1,244	4.2	18	2,000
1907	1	100 / 150 = 0.67	1,178	4.5	22	2,400
1908	1	100 / 250 = 0.4	1,963	5	42	2,400

*While listed as "Peak Power" the figures quoted are probably "Rated Power" at "Rated RPM", short of the true top of the power curve for mechanical reasons, which was typical in the early years.

The 1908 S & N engine is shown on Figs. SO3A and SO3B in "Significant Other" engines.

Sub-Note B

Sizaire et Naudin data compared with Lanchester's proposed Rating Rules

Date	$\frac{PP}{PA}$ HP/cm^2	Lanchester's Rating proposals: $\frac{PP}{PA} \times \sqrt{\left(\frac{B}{S}\right)}$	$\frac{PP}{PA} \times \sqrt{\left(\frac{B}{S} + 0.3\right)}$	$\frac{PP}{PA} \times \left(\frac{B}{S}\right)$
1906	0.16 Datum	0.167 Datum	0.189 Datum	0.175 Datum
1907	0.28 x 1.75	0.229 x 1.37	0.275 x 1.46	0.187 ÷ 1.035* = 0.18 x 1.03
1908	0.53 x 3.31	0.335 x 2.00	0.443 x 2.34	0.212 ÷ 1.087* = 0.195 x 1.11

*Adjusted by Air Standard Efficiency = $\left(1 - \frac{1}{R^{0.4}}\right)$ to value of R = 4.2 as for 1906.

"*Corrected* Mean Piston Speed"

Since PP is proportional to BMEP x PA x MPS, *if* Lanchester's basic proposal of

(PP/PA) x **√**(B/S) = constant was correct, then

BMEP x [**√**(B/S) x MPS] would also be constant.

Although Lanchester himself did not identify this conclusion, others have done so subsequently and called [**√**(B/S) x MPS] "*Corrected* Mean Piston Speed".

As shown above from the Sizaire et Naudin data produced in contemporary circumstances, the basic Lanchester idea, i.e., the adjustment of (PP/PA) by **√**(B/S) *did not yield a constant.* The conception of "*Corrected* Mean Piston Speed" is therefore unsound. It also has an implicit assumption that Piston Height is proportional to Bore and this has not been the case since WW1. This subject is discussed in detail in Note 13 Part I.

Note 36

Dimensions as designed and as cast

The Cosworth DFV designer, Keith Duckworth, wrote about his 1970 cast cylinder heads (58 years after Daimler's 1st fabricated head):-

"…we sectioned a damaged head. There were supposed to be water passages on either side of the sparking plug [boss] *but, in fact, there was solid metal all around it. Either the passages had been too small to cast or the foundry people had chosen to ignore them"* (60).

The solution to improve cooling was to cut out the unwanted cast material and glue in plug access wells – a minor adoption of fabrication.

Note 37

Determination of Thermal and Volumetric Efficiencies

Thermal Efficiency

Given the Specific Fuel Consumption (SFC) of an engine, the Thermal Efficiency (ThE) can be found from:-

$$\text{ThE} = \left(\frac{\text{Power}}{\text{Mechanical Equivalent of Heat Value of Fuel Flow Rate}}\right)$$

Where J = Mechanical equivalent of Heat;

C = Heat Value (Lower Calorific Value + Latent Heat of Evaporation) of Fuel per unit mass;

and $\text{SFC} = \left(\frac{\text{Fuel Mass Flow Rate}}{\text{Power}}\right)$;

then $\text{ThE} = \left(\frac{1}{J} \times \frac{1}{C} \times \frac{1}{\text{SFC}}\right)$

In Imperial units*:-

$$J = 778.26 \text{ ft. Lb. Wt./BTU} = \left(\frac{1}{2544} \; \frac{\text{BHP.Hour}}{\text{BTU}}\right)$$

C is in BTU/lb

SFC is in lb/BHP.Hour

$$\text{So ThE} = \left(2544 \times \frac{1}{C} \times \frac{1}{\text{SFC}}\right)$$

---

*Retained because much of the fuel data was published originally in Imperial units egs. (52, 242, 294, 594).

Alternatively:-

$$\text{ThE} = \left(\frac{\text{Power}}{\text{Ideal Power for the same airflow}}\right)$$

So, as shown in the Power equation of Note 10 and with the same symbols:-

$$\text{ThE} = \left(\frac{\text{Ideal MEP} \times \text{MDR} \times V \times N \times \text{EV} \times \text{ASE} \times \text{EC} \times \text{EM}}{\text{Ideal MEP} \times \text{MDR} \times V \times N \times \text{EV}}\right)$$

So $\text{ThE} = [\,\text{ASE} \times \text{EC} \times \text{EM}\,]$

and $\text{ThE} = \left(2544 \times \frac{1}{C} \times \frac{1}{\text{SFC}}\right)$ as shown above

So $[\text{ASE} \times \text{EC} \times \text{EM}] = \left(2544 \times \frac{1}{C} \times \frac{1}{\text{SFC}}\right)$

Volumetric Efficiency

Therefore, substituting the latter relation for [ASE x EC x EM] in the Power equation 3 in Note 10 and using the same units, i.e.:-

Power in BHP; V in cc; N in RPM;
and with C in BTU/lb; SFC in lb/BHP.Hour as before:-

$$\text{Volumetric Efficiency, EV} = 9.257 \times \left(\frac{\text{C x BHP x SFC}}{\text{MDR x V x N}}\right)$$

Example

Eg.6 1914 Mercedes M93654. DASO 468 p.59.

This is the first example in this review for which full Power and SFC data are available, on a facsimile of the original Daimler test chart **(see P.3).**

$$V = 4{,}483\text{cc}$$

$$PP = \left(\frac{105\text{PS}}{1.01387}\right) = 103.6\text{BHP} \quad @ \quad 3{,}100\text{RPM}$$

$$\text{Corresponding SFC} = 265\text{g/PS.Std} = \left(\frac{265}{1{,}000} \times 2.2046 \times 1.01387\right)$$

$$= 0.592 \text{ lb/BHP.Hour}$$

Fuel:- 50% Petrol + 50% Benzole (from quoted Specific Gravity),

Therefore $C = ½. [(19{,}000 + 135) + (17{,}300 + 169)]$ BTU/lb

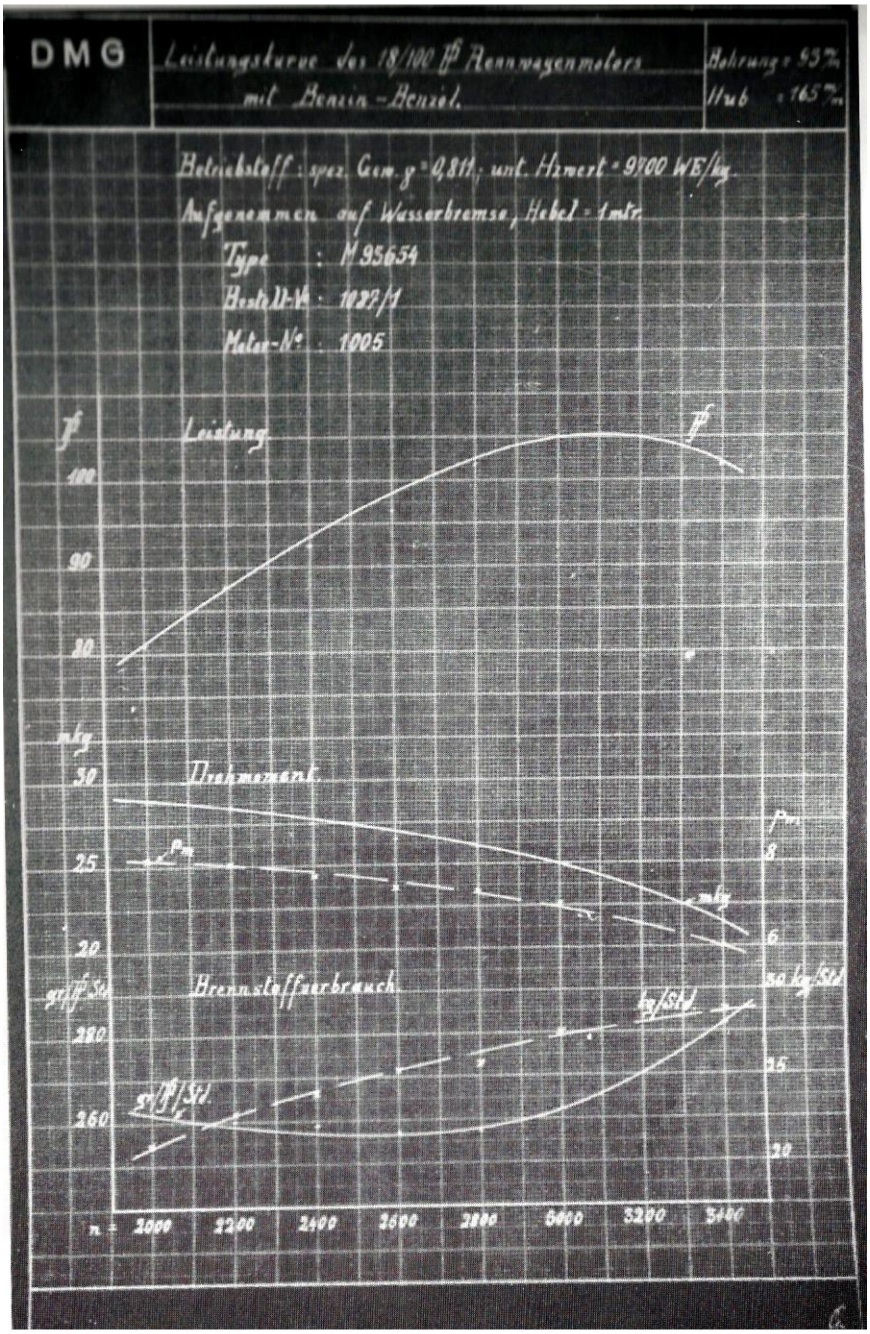
$= 18{,}302$ BTU/lb.

So $\text{BThE} = \left(2544 \times \frac{1}{18{,}302} \times \frac{1}{0.592}\right)$ (BThE = Brake Thermal Efficiency)

$$\text{BThE} = 0.235$$

And $\text{EV} = 9.257 \times \left[\frac{18{,}302 \times 103.6 \times 0.592}{1 \times 4{,}483 \times 3{,}100}\right]$

$$\text{EV} = 0.748.$$


DMG
Leistungskurve des 18/100 PS Rennwagenmotors
mit Benzin-Benzol.
Bohrung = 93 %m
Hub = 165 %m
Betriebstoff: spez. Gew. g = 0,811; unt. Hzwert = 9700 WE/kg.
Aufgenommen auf Wasserbremse, Hebel = 1 mtr.
Type : M 95654
Bestell-No : 1027/1
Motor-No : 1005
Leistung
Drehmoment
Brennstoffverbrauch
mkg
kg/Std
gr/PS/Std
n = 2000 2200 2400 2600 2800 3000 3200 3400

Note 38

FIAT racing aero engines

Although FIAT built an unsuccessful experimental 2 x IL6 1.5 L 2-stroke engine in 1925 (see Note 24) and then a 2 x IL6 1.5 L which won a very short race at Monza in 1927 (66), much of their attention for the 12 years after 1924 was devoted to aero engines for the seaplane Schneider Trophy races and for the World's Air Speed Record (WSR). They powered the Macchi floatplanes which won the former in 1926 (type AS2 60V12 31.3 L) and secured the latter in 1928 (type AS3 60V12 35.2 L).

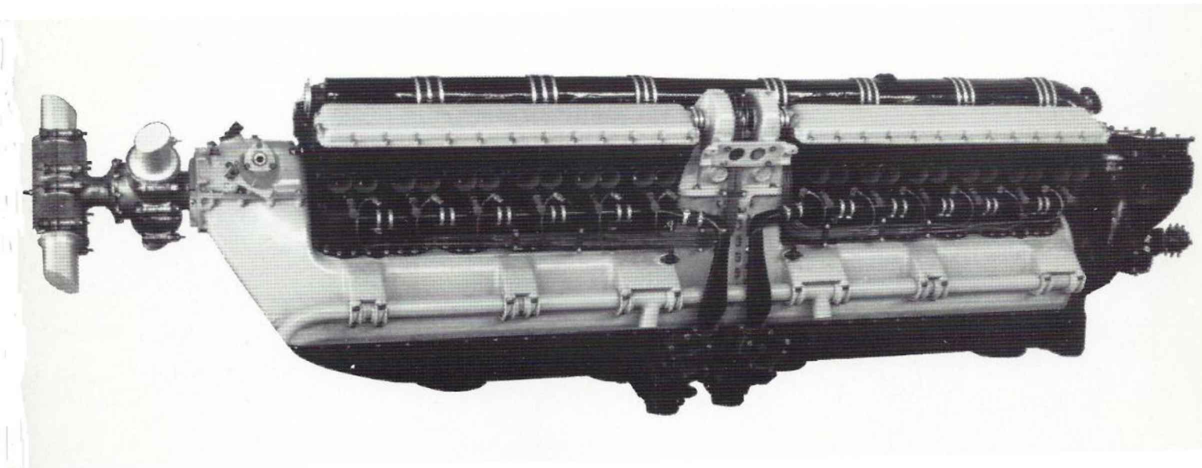
Tranquillo Zerbi, the engineer who had conceived the 2-stroke (66), produced the final successful WSR engine in 1934. This was type AS6 60V24 50.2 L of 3,000 HP (686). The speed of the Macchi-Castoldi type 72 with this engine was 441 MPH which remains unbeaten as the seaplane record.

Principal Characteristics of Fiat Race Engines

Type	Power (hp)	Revolutions per Minute (rpm)	Bore and Stroke (mm)	Capacity (liters)	Compression Ratio	Supercharge Pressure (meters of water)	Weight (kg)
AS2	800	2,300	140 x 170	31.34	6.0	–	388
AS3	1,000	2,400	145 x 175	35.16	6.7	–	422
AS5	1,000	3,200	138 x 140	25.10	8.0	–	345
AS6 (Bleriot Cup)	2,500	3,200	138 x 140	50.20	7.0	4.30	930
AS6 (Speed Record)	3,000	3,300	138 x 140	50.20	7.0	8	930

4.3 mH20 = 42.3 inHg(abs) = +6.25 psi(gauge)
8.0 mH20 = 53.1 inHg(abs) = +11.50 psi(gauge)

1934 FIAT AS6



Credit for data and picture; *Torque Meter* Fall 2007 article by Jerry Wells

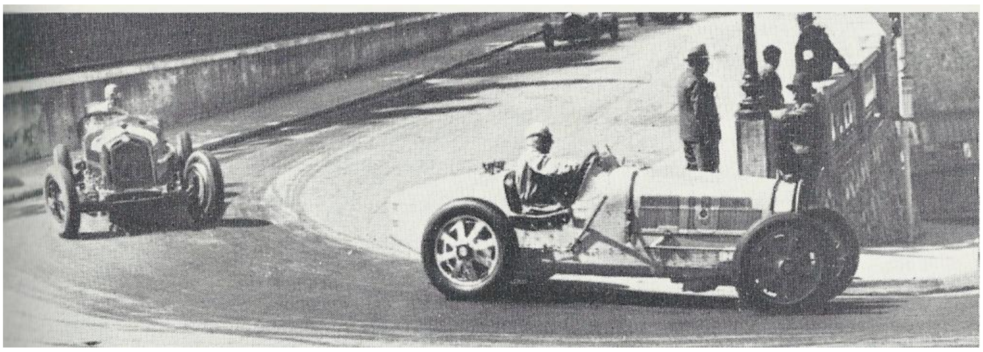
Note 39

Bugatti T51 Overspeed Capability:- Varzi v. Nuvolari at Monaco, 1933

In the famous race-long duel between Achille Varzi (in a Bugatti T51) and Tazio Nuvolari (Alfa Romeo 8C 2.6 L modified Monza) at Monaco in 1933, it is reported that the former over-revved to 7,000 RPM in 3rd gear up the hill to the Casino on the 100th and last lap to beat his rival Italian (887), i.e. 1,500 beyond peak power. The Alfa was also grossly over-revved at the same time and suffered a broken oil pipe – probably from a resonance – (which caused a brief under-bonnet fire). This was followed by a plain main bearing failure when oil pressure dropped (these conclusions are deduced from 3 reports (923,924,925) consolidated logically – another report, considered to be less likely, mentions a broken piston (887) – pity the technical analyst!). Nuvolari stopped, started to push but a mechanic helped and by the rules the driver was then disqualified.

With R = 6 and only 14^0 of overlap the T51 obviously did not suffer immediately-fatal off-cam valve strikes against the pistons or each other, despite the abuse which it suffered. Conversely, had the Alfa been fitted with roller bearings it might have survived oil starvation long enough to finish.

Fig.N39A
Varzi leading Nuvolari at the Station Hairpin
DASO 887



Note 40

The Bugatti change of cylinder head, 1931

	35B	51	
Type	35B	51	
1st raced	1926	1931	
Both IL8 60/100 = 0.6; R = 6; IVP = 1.68 ATA; Elcosine fuel			
Data sources	4, 28, 516	26, 28	
Valve Arrangement			
VIA	0	96^0	
No. per cyl.	2 Inlet 1 Exhaust	2	
IVA/PA	0.36	0.28	
Combustion Chamber			
Shape	Cylindrical	Hemispherical	
Piston Crown	Flat	Flat	
Plug Location	1 under inlets	1 central	
Max. Flame Travel mm	60	45	-25%
Valve Gear			
	SOHC With levers	DOHC With finger followers	
Valve Timing^0			
IO/IC//EO/EC	10/35//50/20	7/40//40/7	
IOD//EOD (OL)	225//250 (30)	227//227 (14)*	-23 EOD (-16 OL)
PP HP	147	185	+26%
@ NP RPM	5,200	5,500	+6%
MGVP m/s	48.0	64.4	+34%
BMPA Bar	13.8	16.4	+19%
ECOM	37.8%	45.0%	+7.2% points
MPSP m/s	17.3	18.3	+ 6%
MVSP m/s	2.22	2.70	+21.6%
Weight**		Small increase for 2nd camshaft and its drive	

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*Miller 1.5 L: 5/38//35/8
223//223 (13)

**No Bugatti engine weights have been found in the literature and none have been obtainable from people concerned with Bugatti engines still in use.

Note 41

Larger-capacity non-CoY engines, 1925 – 1933

While making the point in Eg. 19 regarding increased swept volume (V) of “regular” racing cars over 1928 – 1933 it may be noted that in some *Formule Libre* races over 1925 – 1933 much larger-engined cars were used.

In 1925 Sunbeam raced a 60V12 4L built-up essentially from 2 blocks of their Grand Prix engine driving on one crank (see Fig. N41A).

Maserati in 1929 put 2 of their 1926 IL8 2L engines side-by-side in a joint crankcase (the “*Sedici Cylindri*”) and followed this in 1932 with a double IL8 2.5L version.

Alfa Romeo in 1931 copied Maserati with 2 IL6 1.75L sports engines alongside each other in one *Monoposto* chassis.

Bugatti, whose T50 IL8 5L production engine was his 1st with DOHC, made a racing version, T54, in 1931 (see Fig. N41B).

Fig. N41A
1926 Sunbeam *Formule Libre*
60V12 MSC 67/94 = 0.713 3,977 cc
299 HP @ 5,000 RPM on 60/40 Petrol/Benzole
Showing the modified installation of 2 Roots-type superchargers in parallel (which pre-dated the same arrangement on the 3L V12 Mercedes-Benz M154 by 12 years)
which replaced the original single unit when the latter’s casing cracked several times.
DASO 24

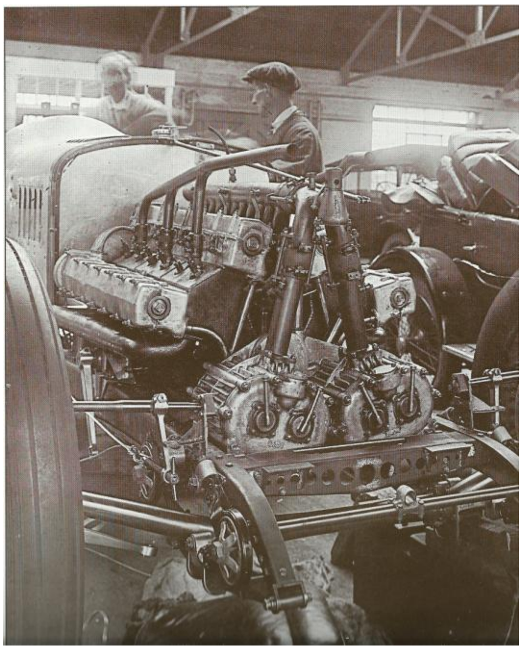


Fig. N41B
1931 Bugatti T54
IL8 MSC 86/107 = 0.804 4,972 cc
As the road-going T50, the 1st DOHC engine built by Bugatti,
the power was 200 HP @ 4,000 RPM on petrol (DASO 308).

Credit; www.wallpaperup.com

Note 42

Comparison between M218 and M25A

Type	M218	M25A	
Data Sources	468	4, 468	
1st raced	1924	1935	
Configuration	IL8	IL8	
Bmm/ Smm	61.7/82.8	78/88	
	= 0.75	= 0.89	
V cc	1,981	3.364	
R	5	7.5	
IVP ATA	1.97	1.66	
Supercharging system	Suction carburetter	Pressurised carburetter	
Fuel	Assumed 50 petrol 50 benzole	50 petrol 50 benzole	
MDR	1.50	1.39	
No. of valves per cylinder	4	4	
VIA^0	Assumed 60	60	
IVD /IVL mm	n.a.	34/8.5	
IVA/PA	n.a.	0.38	
Valve gear	DOHC	DOHC	
IOD^0	n.a.	250	
OL^0	n.a.	45	
PP BHP	(170 PS) 168	(314 PS) 310	
@ NP RPM	7,000	5,800	
BMPP Bar	10.84	14.22	
(BMPA/MDR) Adj. Bar*	9.59	11.64	+21.4%
ECOM %	40.1	48.6	+ 8.5 %points
[(PPA/Vlitres)/MDR]	75.0	75.5	+0.7%
Adjusted HP per Litre per unit Manifold Density Ratio			
MPSP m/s	19.32	17.01	-12%
MGVP m/s	n.a.	44.77	
MVSP m/s	n.a.	2.37	
W Kg	n.a.	203	

*See Appendix 1, Key to abbreviations, Row 79 for explanation of BMPA.

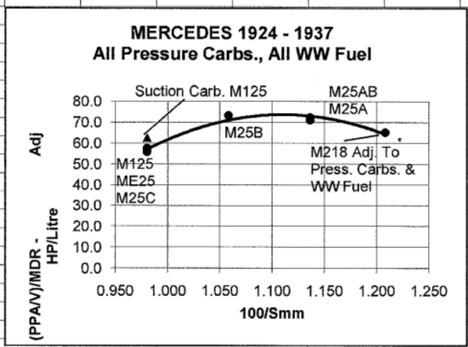
Note 43.
Mercedes-Benz racing engines, 1924 - 1937

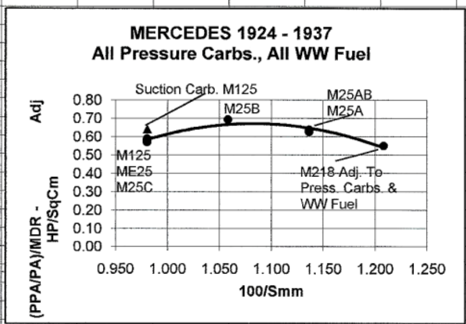
	A	B	C	D	E	F	G	H	I	J	K	L
1	PISTON ENGINE PERFORMANCE	11 January 2009 MERC. 1924-1937										
2	ENGINE IDENTITY											
3	PEP Serial No.											
4	Data Source Ref. File DASO	468	4,468		4,468	4,468	4,468	4,468	4,468	4,468	4,468	
5	YEAR	1924	1934		1934	1934.6	1934.7	1935	1936	1937	1937.5	
6	Make	MERC.	MERC.		MERC.	MERC.	MERC.	MERC.	MERC.	MERC.	MERC.	
7	Model	M218	M25A		M25A	M25AB	M25B	M25C	ME25	M125	M125	
8	Swept Volume Litres	2	3.4		3.4	3.7	4	4.3	4.7	5.7	5.7	
9	Induction System	PC/Suctn	PC/Press		PC/Press	PC/Press	PC/Press	PC/Press	PC/Press	PC/Press	PC/Suctn	
10	Class	RR	RR		RR	RR	RR	RR	RR	RR	RR	
11	GEOMETRY											
12	Configuration	IL8	IL8		IL8	IL8	IL8	IL8	IL8	IL8	IL8	
13	No. of Cylinders CN	8	8		8	8	8	8	8	8	8	
14	No.Cyls/Intake CNI	8	8		8	8	8	8	8	8	8	
15	In. & Ex. Configuration	RSC/CF	RSC/CF		RSC/CF	RSC/CF	RSC/CF	RSC/CF	RSC/CF	RSC/CF	RSC/CF	
16	Comb. Ch'b'r/Piston Config'n	PR/F	PR/LH		PR/LH	PR/LH	PR/LH	PR/LH	PR/LH	PR/MH	PR/MH	
17	Compression Ratio R	5	7.5		7.5	7.5	7.33	8.17	8.2	8.9	8.9	
18	BORE B mm	61.7	78		78	82	82	82	86	94	94	
19	STROKE S "	82.8	88		88	88	94.5	102	102	102	102	
20	Valve Opening/Return System	DOHC	DOHC		DOHC	DOHC	DOHC	DOHC	DOHC	DOHC	DOHC	
21	Valve No./Cyl.-In. VNI	2	2		2	2	2	2	2	2	2	
22	" " " -Ex. VNE	2	2		2	2	2	2	2	2	2	
23	Valve Incl. Angle VIA Deg	60	60		60	60	60	60	60	70	70	
24	Inlet Valve Dia. IVD mm		34		34	35.5	35.5	35.5	37	39	39	
25	Inlet Valve Lift IVL "		8.5		8.5	8.5	8.5	8.5	8.5	8.5	8.5	
26	Inlet Tract Length LIN "											
27	Timing-In. Open IVO Deg		25		25			20	20	15	15	
28	" " Close IVC "		45		45			40	35	42	42	
29	" Ex Open EVO "		50		50			28	27	30	30	
30	" " Close EVC "		20		20			6	10	3	3	
31	In. Open Duration IOD "		250		250	250	250	240	235	237	237	
32	Ex. " " EOD "		250		250			214	217	213	213	
33	In.-Ex. Overlap OL "		45		45			26	30	18	18	
34	Main Journal Dia. MJ mm		63		63			63		67	67	
35	Crank Pin Dia. CP "		53		53			59		66	66	
36	Gudgeon Pin Dia. GP "		22		22			22		25	25	
37	Con. Rod Length CRL "		161		161			168		167	167	
38	Piston Height PH "		94		94			94		96	96	
39	Piston Skirt Length PSL "		74		74			74		81	81	
40	Equiv. PSL - EPSL "		74		74			74		81	81	
41	INFLOW CONDITIONS											
42	Fuel Type	P/B	P/B		A/WW	A/WW	A/WW	A/WW	A/WW	A/WW	A/WW	
43	Fuel Adj. to Petrol AA	1	1		1	1	1	1	1	1	1	
44	Press. @ In. Valve IVP ATA	1.97	1.66		1.66	1.66	1.66	2.1	2	1.9	1.77	
45	Manifold Density Ratio = MDR	1.5	1.39		1.66	1.66	1.66	2.1	2	1.9	1.77	
46	CODE											
47	Induction Code	B	B		B	B	B	B	B	B	B	
48	PERFORMANCE											
49	Peak (Rated) Power PP HP	168	310		349	393	424	456	487	574	580	
50	Crank RPM @ PP NP	7000	5800		5800	5800	5800	5800	5800	5800	5800	
51	Peak Torque TP LbFt											
52	Crank RPM @ TP NT											
53	GEOMETRIC ANALYSIS											
54	B/S	0.745	0.886		0.886	0.932	0.868	0.804	0.843	0.922	0.922	
55	PA SqCm	239.19	382.27		382.27	422.48	422.48	422.48	464.70	555.18	555.18	
56	V/CN cc per cylinder	247.6	420.5		420.5	464.7	499.1	538.7	592.5	707.9	707.9	
57	V cc	1980.5	3364.0		3364.0	3717.8	3992.4	4309.3	4740.0	5662.9	5662.9	
58	IVA SqCm		145.3		145.3	158.4	158.4	158.4	172.0	191.1	191.1	
59	IVA/PA		0.380		0.380	0.375	0.375	0.375	0.370	0.344	0.344	
60	IVL/IVD		0.25		0.25	0.239	0.239	0.239	0.230	0.218	0.218	
61	ISA SqCm		145.3		145.3	151.7	151.7	151.7	158.1	166.6	166.6	
62	ISA/PA		0.380		0.380	0.359	0.359	0.359	0.340	0.300	0.300	
63	MJ/S %		71.6		71.6	0.0	0.0	61.8	0.0	65.7	65.7	
64	CP/S %		60.2		60.2	0.0	0.0	57.8	0.0	64.7	64.7	
65	GP/S %		25.0		25.0	0.0	0.0	21.6	0.0	24.5	24.5	
66	CRL/S		1.83		1.83	0.00	0.00	1.65	0.00	1.64	1.64	
67	B/PH		0.83		0.83			0.87		0.98	0.98	
68	100/Smm	1.208	1.136		1.136	1.136	1.058	0.980	0.980	0.980	0.980	
69	R*VIA	300.0	450.0		450.0	450.0	439.8	490.2	492.0	623.0	623.0	
70	PERFORMANCE ANALYSIS											
71	PP/V=SP HP/Litre	84.8	92.2		103.7	105.7	106.2	105.8	102.7	101.4	102.4	
72	F= (NP-NT)/NP %											
73	MPSP = 2*S*NP m/s	19.32	17.01		17.01	17.01	18.27	19.72	19.72	19.72	19.72	
74	BMPP Bar	10.84	14.22		16.01	16.31	16.39	16.33	15.85	15.64	15.80	
75	MPST m/s											
76	BMTP Bar											
77	RA =0.63/(1-1/R^0.4)	1.327	1.138		1.138	1.138	1.147	1.108	1.107	1.081	1.081	
78	PPA = PP*RA/AA HP	222.9	352.9		397.3	447.4	486.3	505.4	539.1	620.3	626.8	
79	BMPA= BMPP*RA/AA Bar	14.39	16.18		18.22	18.57	18.79	18.09	17.55	16.90	17.08	
80	BMPA/MDR Adj.Bar	9.59	11.64		10.98	11.18	11.32	8.62	8.77	8.89	9.65	
81	TPA = TP*RA/AA Lb.Ft											
82	BMTA =BMTP*RA/AA Bar											
83	PPA/PA HP/SqCm	0.93	0.92		1.04	1.06	1.15	1.20	1.16	1.12	1.13	
84	(PPA/PA)/MDR Adj.HP/SqCm	0.62	0.66		0.63	0.64	0.69	0.57	0.58	0.59	0.64	
	E COM %	40.0	48.6		45.9	46.7	47.3	36.0	36.6	37.1	40.3	

Continued below.

Note 43 continurd

	A	B	C	D	E	F	G	H	I	J	K	L
5	YEAR	1924	1934		1934	1934.6	1934.7	1935	1936	1937	1937.5	
6	Make	MERC.	MERC.		MERC.	MERC.	MERC.	MERC.	MERC.	MERC.	MERC.	
7	Model	M218	M25A		M25A	M25AB	M25B	M25C	ME25	M125	M125	
85												
86	(PPA/PA)*(B/S)/ MDR	0.463	0.589		0.555	0.594	0.602	0.458	0.489	0.542	0.588	
87	PPA/V HP/Litre	112.6	104.9		118.1	120.3	121.8	117.3	113.7	109.5	110.7	
88	(PPA/ V)/ MDR) Adj.HP/Litre	75.0	75.5		71.1	72.5	73.4	55.8	56.9	57.7	62.5	
89	PPA/IVA HP/SqCm		2.43		2.73	2.82	3.07	3.19	3.13	3.25	3.28	
90	PPA/ISA " "		2.43		2.73	2.95	3.21	3.33	3.41	3.72	3.76	
91	MGVP = MPSP*PA/IVA m/s		44.77		44.77	45.39	48.74	52.61	53.27	57.28	57.28	
92	MSVP = MPSP*PA/ISA "		44.77		44.77	47.39	50.89	54.93	57.97	65.70	65.70	
93	BNP = B*NP "	7.20	7.54		7.54	7.93	7.93	7.93	8.31	9.09	9.09	
94	MVS = IVL*NP/(83.333*IOD) "		2.37		2.37	2.37	2.37	2.47	2.52	2.50	2.50	
95	MPD @ nom'l (CRL/S)=2 g	2834.5	2068.2		2068.2	2068.2	2220.9	2397.2	2397.2	2397.2	2397.2	
96	MPD @ actual CRL g											
97	MOD. MPSP (MMPSP) m/s											
98	(NPx(MPSP)^2)/10^5	26.13	16.79		16.79	16.79	19.36	22.55	22.55	22.55	22.55	
99	KF1 for FPMEP	0.75	0.75		0.75	0.75	0.75	0.75	0.75	0.75	0.75	
100	KF2 for FPMEP*10^7	9	9		9	9	9	9	9	9	9	
101	EIMPA Bar	18.50	18.76		20.79	21.14	21.65	21.17	20.63	19.90	20.08	
102	Estd. Mech. Effy. EEM %	77.8	86.3		87.6	87.8	86.8	85.4	85.1	84.9	85.0	
103	EIMPA/MDR Bar	12.34	13.50		12.53	12.73	13.04	10.08	10.31	10.48	11.34	
104	EIMPA/(MDR*(MPSP)^0.5)= **SPPA**	2.81	3.27		3.04	3.09	3.05	2.27	2.32	2.36	2.55	
105												
106	**SPPB**	2.39	2.78		2.54	2.59	2.56	1.77	1.82	1.89	2.08	
107	**SPPB** -f(CRL/S)=**SPPC**	2.04	2.82		2.58	2.24	2.21	1.96	1.47	2.09	2.28	
108	**Delta from 3*(B/PH)^1/3 %**		-0.1		-8.4			-31.6		-30.0	-23.4	
109												
110	EBMTA	15.34	17.06		19.06	19.41	19.64	18.93	18.39	17.75	17.93	
111	**Delta EBMTA Act from Est %**											
112												
113												
114	**SPEED CORRn FACTOR - SCF**		159.89		159.89	156.03	149.00	153.42	147.31	144.46	142.77	
115	NP Repeat - RPM		5800		5800	5800	5800	5800	5800	5800	5800	
116	GS = Actual NP/SCF		36.3		36.3	37.2	38.9	37.8	39.4	40.1	40.6	
117	KS = 47.4 or 38.6		38.6		38.6	38.6	38.6	38.6	38.6	38.6	38.6	
118	**Delta Actual from KSxSCF %**		-6.0%		-6.0%	-3.7%	0.8%	-2.1%	2.0%	4.0%	5.2%	
119												
120												
121	WEIGHT - W - kg		203		203	203	206	215	211	223	223	
122	**PP/W - HP/kg**		1.53		1.72	1.94	2.06	2.12	2.31	2.57	2.60	
123	RFW - Litres adj.	3.76	5.37		5.37	5.64	6.51	7.58	7.95	8.69	8.69	





	A	B
149		M218
150		Adj.
151		to Press
152		Carbs.,
153		WW Fuel
154	100/Smm	1.208
155	(PPA/PA)/MDR Adj HP/SqCm	0.55
156	(PPA/V)/MDR Adj HP/Litre	65.2

Note 44
Auto Union racing engines, 1934 – 1937

	A	B	C	D	E	F	G
1	PISTON ENGINE PERFORMANCE	11 January 2009 AUTO UNION 1934-1937					
2	ENGINE IDENTITY						
3	PEP Serial No.						
4	Data Source Ref. File DASO	4,468	4,30,276, 382	4,30,276, 382	4,30,276, 381,382	4,30,276, 382,711	4,30,276, 382,711
5	YEAR	1934	1934	1935	1935.5	1936	1937
6	Make	MERC.	A UNION	A UNION	A UNION	A UNION	A UNION
7	Model	M25A	A	B	B/C	C	R
8	Swept Volume Litres	3.4	4.4	4.9	5.6	6	6.3
9	Induction System	PC/Press	PC	PC	PC	PC	PC
10	Class	RR	RR	RR	RR	RR	RR
11	GEOMETRY						
12	Configuration	IL8	45V16	45V16	45V16	45V16	45V16
13	No. of Cylinders CN	8	16	16	16	16	16
14	No.Cyls/Intake CNI	8	16	16	16	16	16
15	In. & Ex. Configuration	RSC/CF	RSC/CF	RSC/CF	RSC/CF	RSC/CF	RSC/CF
16	Comb. Ch'b'r/Piston Config'n	PR/LH	H/F	H/MH	H/MH	H/MH	H/MH
17	Compression Ratio R	7.5	7	8.95		9.2	9.2
18	BORE B mm	78	68	72.5	72.5	75	77
19	STROKE S "	88	75	75	85	85	85
20	Valve Opening/Return System	DOHC	SOHC/PR	SOHC/PR	SOHC/PR	SOHC/PR	SOHC/PR
21	Valve No./Cyl.-In. VNI	2	1	1	1	1	1
22	" " " -Ex. VNE	2	1	1	1	1	1
23	Valve Incl. Angle VIA Deg	60	90	90	90	90	90
24	Inlet Valve Dia. IVD mm	34	*39*	*39*	39	*39*	*39*
25	Inlet Valve Lift IVL "	8.5	*10*	*10*	10	*10*	*10*
26	Inlet Tract Length LIN "						
27	Timing-In. Open IVO Deg	25					
28	" " Close IVC "	45					
29	" Ex Open EVO "	50					
30	" " Close EVC "	20					
31	In. Open Duration IOD "	250	*260*	*260*	*260*	*260*	*260*
32	Ex. " " EOD "	250	*40*	*40*	*40*	*40*	*40*
33	In.-Ex. Overlap OL "	45					
34	Main Journal Dia. MJ mm	63	62	62	70	70	70
35	Crank Pin Dia. CP "	53	58	58	68	68	68
36	Gudgeon Pin Dia. GP "	22	22	22	22	22	22
37	Con. Rod Length CRL "	161	164	164	168	168	168
38	Piston Height PH "	94					
39	Piston Skirt Length PSL "	74					
40	Equiv. PSL - EPSL "	74					
41	INFLOW CONDITIONS						
42	Fuel Type	A/WW	P/B?	?	?	A	A
43	Fuel Adj. to Petrol AA	1	1	1	1	1	1
44	Press. @ In. Valve IVP ATA	1.66	1.6	1.75		1.95	1.94
45	Manifold Density Ratio = MDR	1.66	*1.36*	*1.63*		*1.7*	*1.7*
46	CODE						
47	Induction Code	B	B	B	B	B	B
48	PERFORMANCE						
49	Peak (Rated) Power PP HP	349	295	375		520	545
50	Crank RPM @ PP NP	5800	4500	4800		5000	5000
51	Peak Torque TP LbFt		380	477		627	650
52	Crank RPM @ TP NT					2500	
53	GEOMETRIC ANALYSIS						
54	B/S	0.886	0.907	0.967	0.853	0.882	0.906
55	PA SqCm	382.27	581.07	660.52	660.52	706.86	745.06
56	V/CN cc per cylinder	420.5	272.4	309.6	350.9	375.5	395.8
57	V cc	3364.0	4358.0	4953.9	5614.4	6008.3	6333.0
58	IVA SqCm	145.3	191.1	191.1	191.1	191.1	191.1
59	IVA/PA	0.380	0.329	0.289	0.289	0.270	0.257
60	IVL/IVD	0.25	0.256	0.256	0.256	0.256	0.256
61	ISA SqCm	145.3	196.0	196.0	196.0	196.0	196.0
62	ISA/PA	0.380	0.337	0.297	0.297	0.277	0.263
63	MJ/S %	71.6	82.7	82.7	82.4	82.4	82.4
64	CP/S %	60.2	77.3	77.3	80.0	80.0	80.0
65	GP/S %	25.0	29.3	29.3	25.9	25.9	25.9
66	CRL/S	1.83	2.19	2.19	1.98	1.98	1.98
67	B/PH	0.83					
68	100/Smm	1.136	1.333	1.333	1.176	1.176	1.176
69	R*VIA	450.0	630.0	805.5		828.0	828.0
70	PERFORMANCE ANALYSIS						
71	PP/V=SP HP/Litre	103.7	67.7	75.7		86.5	86.1
72	F= (NP-NT)/NP %						
73	MPSP = 2*S*NP m/s	17.01	11.25	12.00		14.17	14.17
74	BMPP Bar	16.01	13.46	14.11		15.49	15.40
75	MPST m/s					7.08	
76	BMTP Bar		14.86	16.41		17.78	17.49
77	RA =0.63/(1-1/R^O.4)	1.138	1.165	1.079		1.071	1.071
78	PPA = PP*RA/AA HP	397.3	343.6	404.6		556.7	583.4
79	BMPA= BMPP*RA/AA Bar	18.22	15.68	15.23		16.58	16.49
80	BMPA/MDR Adj.Bar	10.98	11.53	9.34		9.75	9.70
81	TPA = TP*RA/AA Lb.Ft		442.6	514.6		671.2	695.9
82	BMTA =BMTP*RA/AA Bar		17.30	17.70		19.03	18.72
83	PPA/PA HP/SqCm	1.04	0.59	0.61		0.79	0.78
84	(PPA/PA)/MDR Adj.HP/SqCm	0.63	0.43	0.38		0.46	0.46
85	(PPA/PA)*(B/S)/ MDR	0.555	0.394	0.363		0.409	0.417
86	PPA/V HP/Litre	118.1	78.8	81.7		92.7	92.1
87	(PPA/ V)/ MDR) Adj.HP/Litre	71.1	58.0	50.1		54.5	54.2
88	PPA/IVA HP/SqCm	2.73	1.80	2.12		2.91	3.05

Note 44 continued

	A	B	C	D	E	F	G
5	YEAR	1934	1934	1935	1935.5	1936	1937
6	Make	MERC.	A UNION	A UNION	A UNION	A UNION	A UNION
7	Model	M25A	A	B	B/C	C	R
89							
90	PPA/ISA " "	2.73	1.75	2.06		2.84	2.98
91	MGVP = MPSP*PA/IVA m/s	44.77	34.20	41.47		52.39	55.22
92	MSVP = MPSP*PA/ISA "	44.77	33.35	40.43		51.08	53.84
93	BNP = B*NP "	7.54	5.10	5.80		6.25	6.42
94	MVS = IVL*NP/(83.333*IOD) "	2.37	2.08	2.22		2.31	2.31
95	MPD @ nom'l (CRL/S)=2 g	2068.2	1061.0	1207.2		1484.6	1484.6
96	MPD @ actual CRL g						
97	MOD. MPSP (MMPSP) m/s						
98	(NPx(MPSP)^2)/10^5	16.79	5.70	6.91		10.03	10.03
99	KF1 for FPMEP	0.75	0.75	0.75	0.75	0.75	0.75
100	KF2 for FPMEP*10^7	9	9	9	9	9	9
101	EIMPA Bar	20.79	17.15	16.71		18.35	18.26
102	Estd. Mech. Effy. EEM %	87.6	91.4	91.1		90.4	90.3
103	EIMPA/MDR Bar	12.53	12.61	10.25		10.80	10.74
104	EIMPA/(MDR*(MPSP)^0.5)= **SPPA**	3.04	3.76	2.96		2.87	2.85
105							
106	**SPPB**	2.54	3.29	2.65		2.58	2.57
107	**SPPB** -f(CRL/S)=**SPPC**	2.58	3.05	2.40		2.49	2.48
108	**Delta from 3*(B/PH)^1/3 %**	-8.4					
109							
110	EBMTA	19.06	16.57	16.13		17.44	17.35
111	**Delta EBMTA Act from Est %**						
112							
113							
114	**SPEED CORRn FACTOR - SCF**	159.89	159.29	164.24		152.71	148.42
115	NP Repeat - RPM	5800	4500	4800		5000	5000
116	GS = Actual NP/SCF	36.3	28.3	29.2		32.7	33.7
117	KS = 47.4 or 38.6	38.6	38.6	38.6	38.6	38.6	38.6
118	**Delta Actual from KSxSCF %**	-6.0%	-26.8%	-24.3%		-15.2%	-12.7%
119							
120							
121	WEIGHT - W - kg	203				245	
122	**PP/W - HP/kg**	1.72				2.12	
123	RFW - Litres adj.	5.37	7.63	8.13	10.45	10.81	11.10
124							
125							

Note 44B

Mercedes and Auto Union (Bore/Cylinder-centres) ratio, 1934 – 1937

The gradual increase of the ratio of (Bore/Cylinder-centres) for the rival engines was as follows:-

	Mercedes	
!934	1936	1937
M25A	ME25	M125
78/95 = 0.821	86/95 = 0.905	94/104 = 0.904

Roller main and big-end bearings in all engines.

	Auto Union	
A	C	R
68/85 = 0.800	75/85 = 0.882	77/85 = 0.906
Plain mains & big-ends	Plain mains & roller big-ends	

The block structures were quite different between the two makes, as described in Egs 21, 22 and 23. The Mercedes heads were integral. The gasket for the detachable head of the Auto Union R-type, with a land of only 8mm between bores, had to seal a maximum combustion pressure starting from an inlet charge entering at 1.94 ATA followed by a compression ratio of 9.2.

Note 45

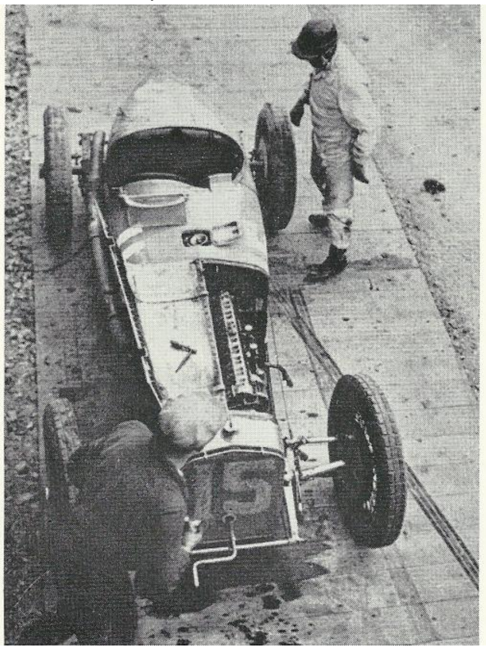
Driver overheating in Mercedes-Benz W154

An interesting - and important – detail of the Mercedes 1938 season was that, 12 years after the 1926 Delage 15-S-8 burned its drivers' feet (see Fig. N45A), the German company had not taken aboard the need for really effective insulation between the engine and driving compartments. This was especially needed with a close-cowled V12 filling most of the former space and an inclined engine axis which then brought the starboard exhaust close to the drivers' right foot.

Seaman's both feet suffered from this defect in winning the German GP (775). In the later Italian GP, in the usual Monza hot weather and when an exhaust gasket leaked as well, Caracciola received a burnt foot. He had to be relieved by Brauchitsch (who had already retired) for several laps (612) (see Fig. N45B). He then continued so as to gain enough points by finishing 3rd to become the Champion of Europe. His injury was sufficient to prevent him competing in the last race at Donington (776). Nuvolari, driving for Auto Union, won both races.

Not surprisingly, Caracciola asked for a better firewall for the 1939 car (468)!

Fig. N45A
1926 Delage
Spanish Grand Prix



DASO 39

Note the exhaust on the driver's side.

Benoist has gone for medical attention and Senéchal has just volunteered to take over. The car was brought home 2nd.

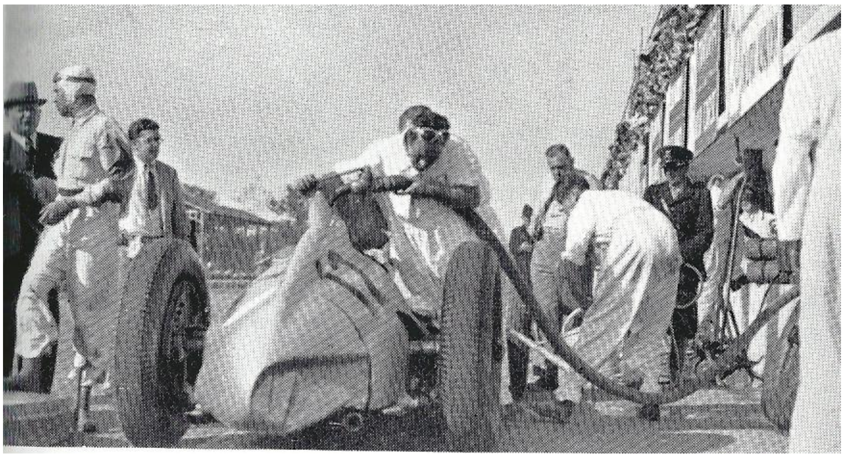


Fig. N45B
Mercedes-Benz
Italian Grand Prix

Carracciola is telling Neubauer (out of picture) and Sailer that he will have to be spelled. Uhlenhaut also on left hand of picture.

The dent in the tail was caused by a spin at the chicane on Lap 2.

Credit; *Grands Prix 1934-1939;* R. Fellowes & R. Walkerley

Note 46

Origin of the Alfa Romeo 158

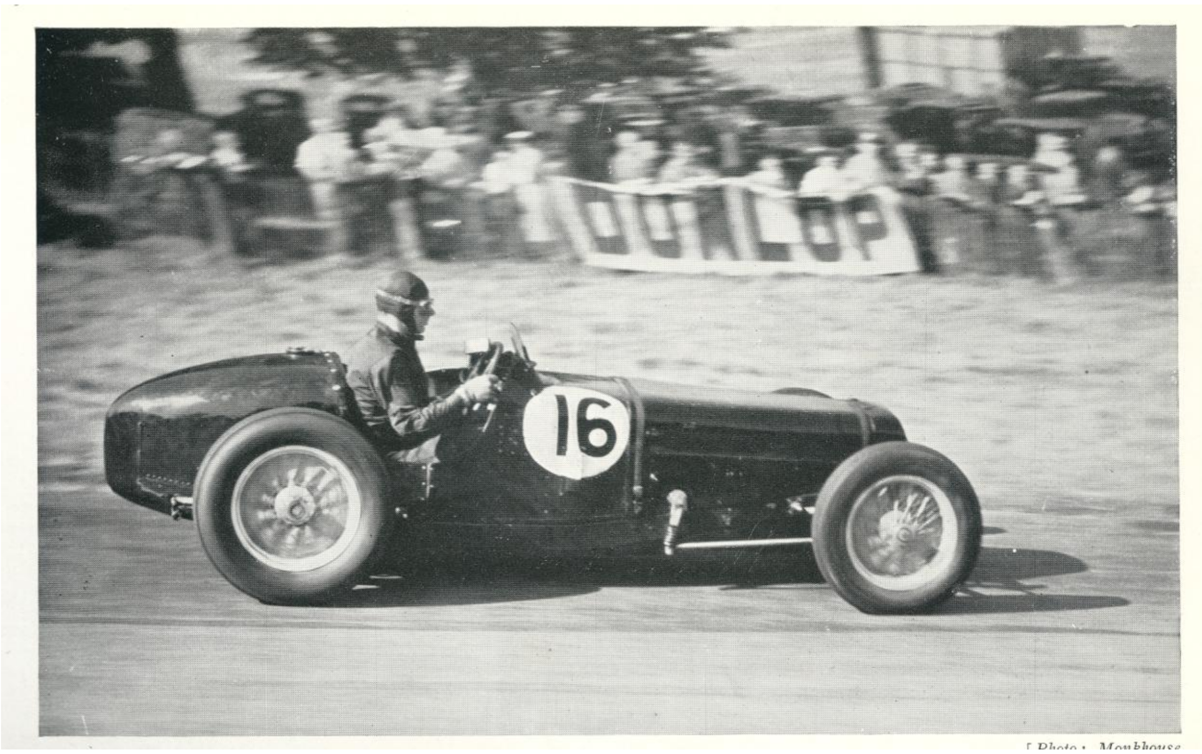


Fig.N46A
1936. Dick Seaman driving the 1927 Delage 15-S-8 rejuvenated by Guilio Ramponi.
DASO 660

What caused Enzo Ferrari to propose to Alfa Romeo in early 1937 the joint project for a new Voiturette which became Tipo 158 (1.5 *Litri, 8 Cylindri)*? Could it have been the successive victories in 1936 at Pescara (*Coppa Acerbo,* 16th August) and Berne (*Prix de Berne*, 23rd August) in the Voiturette races preceding the Grand Prix races at these circuits of the rejuvenated 1927 Delage driven by Dick Seaman?

This IL8 1,5 L car, basically 9 years old but somewhat updated by Guilio Ramponi, had beaten the latest IL6 Voiturettes of Maserari and ERA and Seaman had lapped in practice at 88-89% of the speeds of the 750 Kg formula Grand Prix cars which took the poles for the main races.
(See http://www.kolumbus.fi/leif.snellman).

Ferrari would have been well informed of these events, although probably not himself present at Pescara and certainly not at Berne (after 1934 he scarcely ever attended Italian races and none abroad). He might well have thought that what could be done by an IL8 iron-block engine in a rigid-axled chassis could be improved greatly by a miniaturised 8C-35 having an Al-alloy engine in a chassis with all independent suspension. A successful Alfa Romeo Voiturette would be soothing to his pride after 2½ years of German Grand Prix wins only occasionally interrupted by Tazio Nuvolari's virtuosity. There was also the technical possibility of a 2 x IL8 3 L adaptation which might recoup the Grand Prix situation.

Certainly, a year before the 1938 formula cars began racing, it would have been too soon to have anticipated a 1941 1.5 L GP formula, although the existence and performance of the Alfa 158 must have influenced the AIACR consideration of that 2 years later.

Note 47

The Ferrari SOHC V12 inlet limitation

In the 1957 type 250 Testa Rossa 60V12 3 L Sports-Racing engine, which was B/S = 73mm/58.8 = 1.24, still based on the original type 125 block casting and still SOHC, Ferrari finally fitted a cylinder head with individual inlet ports (138). Each had its own carburetter choke. This improvement in breathing was made possible by moving the single plug per cylinder from the inlet to the exhaust side where they were arranged as in the 1951 type 375 4.5 L 24-plug engine, i.e. the individual exhaust ports being displaced longitudinally to a 1-2-2-1 grouping. The plug points were still deeply recessed from the combustion chamber (see Fig. SO14B which has the 24 plug scheme).

The new engine was claimed to give 306 CV (302 BHP) @ 7,400 RPM (138), representing BMPP = 12.4 Bar on Petrol at R = 9.6; MPSP was 14.5 m/s.

The power claim seemed optimistic at the time but the engine was very successful, winning the Sports Car Championship (limited to 3 L) including Le Mans, in 1958, 1960, and 1961. Significantly, however, it was beaten in the 1959 Championship and at Le Mans by the Aston Martin DBR1 IL6 3 L which had just under 270 BHP*.

If it is accepted that the 250TR claimed power was 10% optimistic, then BMPP = 11.1 Bar. This figure, on Petrol, is still comfortably 10% better than the 1951 24-plug type 375 on an alcohol mixture fuel. It indicates how that engine had been restricted by its breathing.

The 250TR had MGVP = 60 m/s and (R x VIA) = (9.6 x 60^0) = 576^0.

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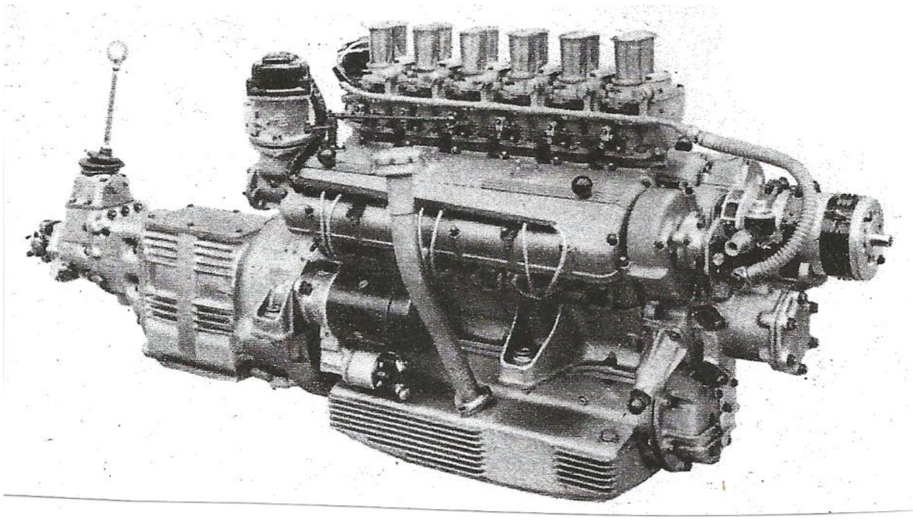
*DASO 267. Racing with the DB Aston Martins. Vol. 2. J. Wyer & C. Nixon. 1980.

Fig. N47A
1957 Ferrari 250TR
Motor 23 April 1958



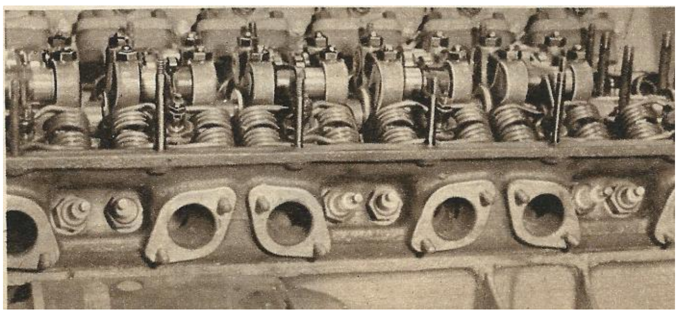


Fig. N47B
Showing details of the 250TR plug arrangement on the exhaust side.
DASO 138

Note 48

"Super-Tuning "with short life

A classic way of "Super-Tuning" to produce a short-time power gain was to reduce the inlet valve/seat contact area virtually to a line. This raised airflow but the seats hammered-out quickly so that valve-tappet clearance was lost and the valve began to leak. Such a modification was described by Tony Rudd as it was applied to the V8 1.5L BRM engines for the 1965 Italian GP, where it raised airflow by 5% (40).In that case Rudd was successful in persuading his drivers (Graham Hill and Jackie Stewart) not to use full RPM initially and so preserve the power gain until signalled to do so towards the end of the race. The 2 BRMs then drew away from Jimmy Clark in a Lotus-Climax, whose engine blew up in trying to hold them and the BRM team took 1st and 2nd places. See Fig. N48.

When good grid positions became of greater importance as aero downforce with the consequential *up*flowing wake made passing much more difficult, the teams introduced special engines to obtain the best Qualification lap time even although they only lasted a few laps. Such engines could make good use of "Super-Tuning". Race-life units were then fitted. (These arrangements came to an end by regulation after the end of this review when in 2003 the same engine had to be used for Qualification and the race, followed in 2004 and onwards by rules forcing engine life to be extended, year by year.)

Although *not* "Super-Tuning", strictly, the advantage of a brand new clean engine for a short period was quoted for the 1962 Coventry Climax FWMV2 in ref. (1071). This gave 182 HP @ 9,000 RPM when newly built but, after a film of black carbon formed on the combustion chamber surfaces with running which prevented heat from being reflected back into the burning charge, the power dropped to 177 HP @ 8,500 RPM, i.e. a loss of 2.7%. It is presumed that this is a universal effect.

Fig. 48
The BRMs at Monza in 1965

Stewart took advantage of a small mistake by Hill on the penultimate lap to pip his team-leader to the finish to win his 1st Grand Prix, going on to accumulate 27 in 99 starts (27%).

Note the BRM understeering.

DASO 938

To return to home page: Home Page

Note 49

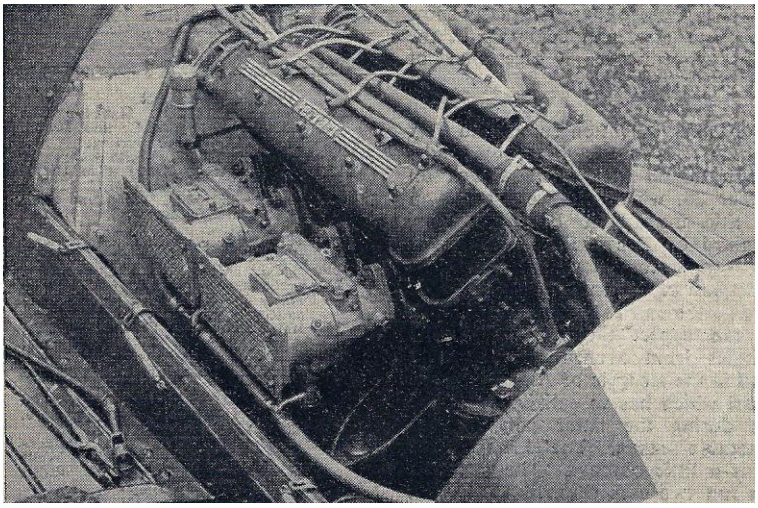
The 2.5L NA Ferrari at Bari in 1951

Enzo Ferrari was so anxious to make a good showing of his new type 625 2.5L NA car in the F1 race at Bari on 2 September 1951 that he broke an agreement with Stirling Moss who, as a rising young driver with 2L HWM Formula 2 and 3.4L Jaguar sports cars, had been invited to drive it on its debut. It would have been a reasonable performance step-up for Moss, well within his proven capability, which is presumably why Ferrari made the offer in the first place. Moss had actually finished 3rd in the 1950 Bari F1 race in the nominal 2-seater 2L HWM, behind 2 Alfa 159s.

Instead, Piero Taruffi, admittedly a driver with engineering training who had good experience of the 4.5L GP car, was given the drive at the last minute after Moss had travelled 1,000 miles to Bari.

Moss was so annoyed by this cavalier treatment that he resolved never to drive a works Ferrari – which unfortunately hurt his career more than it hurt Ferrari (921). This situation might have been changed by a generous personal Ferrari offer in early1962, which followed his two outstanding 1961 victories in an out-of-date Lotus-Climax over the otherwise dominant type 156 Ferraris, but Moss' accident at Easter Goodwood intervened.

Fig.N49A
1951 Ferrari 625 Prototype
IL4 94/90 = 1.044 2,498 cc
200 HP @6,500 RPM on 80/20 Petrol/Alcohol
DASO 8



Motor

To return to home page: [Home Page]

Note 50

The 1954 Formula

Laurence Pomeroy (32) states that he attended the October 1951 FIA meeting which fixed the post-1953 formula (therefore giving over 2 years notice for design and development) and that a “slow-down” mood amongst the other attendees was against prolonging the 1.5L PC/4.5L NA rules scheduled to last until the end of 1953, as desired by Britain and Germany. The former delegates hoped that the V16 PC BRM would come good by that time and the latter that Mercedes-Benz would enter a new V12 PC car (468).

This mood then led the British to suggest 750 cc for PC engines. They hoped that “1/2” of the long-delayed 1.5L BRM could be ready for 1954. This was accepted and, as the 1 PC: 3 NA ratio had not quite brought equality in 1951, an extra 10% was proposed for NA, i.e. 1 PC : 3.3.NA. Hence, rounded-up from 2.475, 2.5L for NA.

The 2.5L Ferrari demonstration at Bari shortly before the meeting (Taruffi lapped at 91% of Fangio’s Alfa 159 speed on pole in practice) probably helped to settle the matter.

Of course it was not known at the meeting that, after Alfa Romeo announced their withdrawal at the end of 1951, Formula 1 1.5L PC/4.5L NA rules would be abandoned by classic 0.race organisers for 1952 – 1953 in favour of 2L NA Formula 2,

To return to home page: Home Page

Note 51

Daimler-Benz and Ferrari

Two firms have been very much to the fore in Grand Prix racing, one – Daimler-Benz – in the first half of the 20th Century; and the other – Ferrari – in the 2nd half.

Daimler and Daimler-Benz

(racing as Mercedes and Mercedes-Benz: 1908 to 1955

Mercedes, and from 1926 Mercedes-Benz, took part in Grand Prix motor-racing directly from 1908 (after participating in pre-Grand Prix events) but by no means continuously. When their absence was not enforced by war and post-war situations they chose their entries carefully for returns in advertising (sometimes *national* advertising as in the 1934 – 1939 period) and in technical advances. Their approach from the earliest Grand Prix days could be expressed by 2 principles:-

(1): their technical approach has been "Conservative Pioneering;
(2): expense has been no objection.

A "pioneer" has been defined by a cynical American as "*A man alone on the prairie, face down, with an arrow in his back!*". However, in engineering "Conservative Pioneering" may be defined as:-

Retaining technical features which have succeeded previously while seeking constantly for novelties which promise improvements which, **if** *they pass rigorous pre-service testing for performance gains with reliability, can take their place.*

Obviously, this is a very costly procedure and here the 2nd principle came into play at Mercedes-Benz up to the end of 1955. This also applied to their racing operations as well as to car design, see Sub-Note A.

Post 1955: cars

Having retired from direct Grand Prix participation with their own cars in 1955 and after considering in 1991 and rejecting a proposal to return in that way (via Sauber), Daimler-Benz from 1992 part-financed specialist chassis constructors using their own names instead:-

Firstly, Sauber (1993 – 1994 seasons;
Subsequently, McLaren (1995 onwards past the end of this review. From January 2000 Daimler (then Daimler-Chrysler until May 2007) owned 40% of the McLaren parent group. The latter bought their stock back by 2011).

Post 1955: engines

In November 1993 Daimler bought 25% 0f the racing engine manufacturer Ilmor and from then the Grand Prix units bore the name Mercedes-Benz. (Post this review Daimler increased their share of the company to 55% in 2002 and to full ownership and re-naming in 2005). The GP engines were supplied to Sauber in 1994 and then to McLaren.

Abandonment of the 2nd principle

To cope with the cost of modern Grand Prix competition even Daimler-Benz had in the more recent years of this review accepted large-scale non-automotive sponsorship as part of racing with McLaren.

[Post this review

In November 2009 Daimler bought most of the ownership of the successful Brawn team. It now enters the cars as Mercedes AMG, made in the UK and with engines supplied by their fully-owned but UK-based company.]

CoY

Mercedes or Mercedes-Benz were CoY in this review in 3 periods separated by long gaps:- 1908, 1914, then 1935, 1937, 1938, 1939, then 1954 and 1955.

After another long gap the Ilmor engine, badged as Mercedes-Benz, powered the McLaren CoY in 1998 and 1999 (Drivers' only).

Ferrari

Ferrari have raced in Grands Prix continuously since 1948 ,a record unapproachable by any other firm. While Enzo Ferrari was alive this devotion was because of his own passion for motor-racing. Many sports car races were also contested. His road cars were produced only to help finance the racing cars, which was the converse of every other company making ordinary cars and competing in races with specialist departments.

The variety of engines built for GPs in the period of this review comprised:-

2, 4, 6, 8, 10 and 12 cylinders;
In Line, Vee and Flat configurations;
Naturally-Aspirated, Mechanically-Supercharged and Turbo-Charged;

a range which no-one else can rival.

CoY

In this review Ferrari were CoY in:-
1949, 1952, 1953, 1956 (using basically Lancia cars), 1958 (Drivers' only), 1961, 1964, 1975, 1976 (Constructors'), 1977, 1979, 1982 (Constructors'), 1983 (Constructors') and then not until 1999 (Constructors') and once again, after a 21 year gap, both Championships in 2000.
[Post the review they have held both titles each year from 2001 to 2004 inclusive and again in 2007.]

Sub-Note A

Daimler:-"Expense no objection" in running racing operations

Apart from practising "Conservative Pioneering" with all its extra expense in their racing car design and manufacture Daimler (and Daimler-Benz after them) applied their principle of "Expense no objection" in the organisation and management of their racing teams. Examples are given below:-

- 1908 French GP – the cars were tried on the Dieppe circuit 4 months before the race in which they finished 1st and 4th;
- 1914 French GP – for this race trials were run at Lyons 3 months before the event for which they entered 5 cars, brought 6 + a spare chassis (1076), and finished 1st, 2nd and 3rd;
- 1924 Targa Florio – practised 3 months before winning the race;
- 1937 – 1939 GPs – entered 4 to 5 cars and had an equal number at the works being overhauled for the next race with engines 2 x the number of chassis. A supercharged lorry was available to rush a spare car to a race if needed;
- 1939 1.5L Tripoli GP – unwilling to let the Italians steal a march on them by converting this race from GP Formula to Voiturette Formula with only 8 months notice, Daimler rushed through 2 cars to suit and took 1st and 2nd places;
- 1939 1.5L Nuffield Trophy – this was a case where Daimler *would* have spent the money to compete but refused an earnest request by their English driver Dick Seaman to send him with a single W165 to compete in this 1.5L race because they could not prepare *both* existing cars post-Tripoli in the time available. Their " Conservative" principle would not risk the failure of a singleton entry (775) (as had happened at Le Mans in 1930 with the SSK when the firm was short of money);
- 1954 – 1955 GPs – built 20 cars, had up to 6 at a race and built another "racing lorry" to back-up the team. In 1955 built 2 cars with a wheelbase 6 cm (2.7%) shorter than that year's medium-length specially for the Monaco GP. Took 5 car types to Monza in 1955 to settle the best for the Italian GP 3 weeks later on the combined road + banked track but still needed the "racing lorry" to bring another hastily-built type from Stuttgart during the meeting. Won and came 2nd;
- 1955 Mille Miglia – Practised 3 months beforehand, entered 4 300SLRs and took 1st and 2nd places;
- 1955 Le Mans – having "Conservatively" kept drum brakes for the 300SLR, despite Jaguar's proof in winning the 1953 race that disc brakes were superior, they demonstrated their "Pioneering" by fitting supplemental air brakes.

Note 52

Fuel Injection

Fuel injection substituted mechanical power drawn from the crank for pneumatic power drawn from the inlet flow by some of the pressure drop across the carburetter choke. Generally it meant that an unobstructed inlet manifold* could be used for maximum power while still retaining adequate fuel/air mixing at lower engine speeds for good acceleration.

In the Bosch system the power was abstracted directly. In the later (1956) Lucas system the pump drew electrical power from a battery which was in turn charged by an alternator driven from the crank. In so far as the battery could be allowed to run down the car had additional energy on board.

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*Depending on the type of throttle used.

To return to home page: Home Page

Note 53

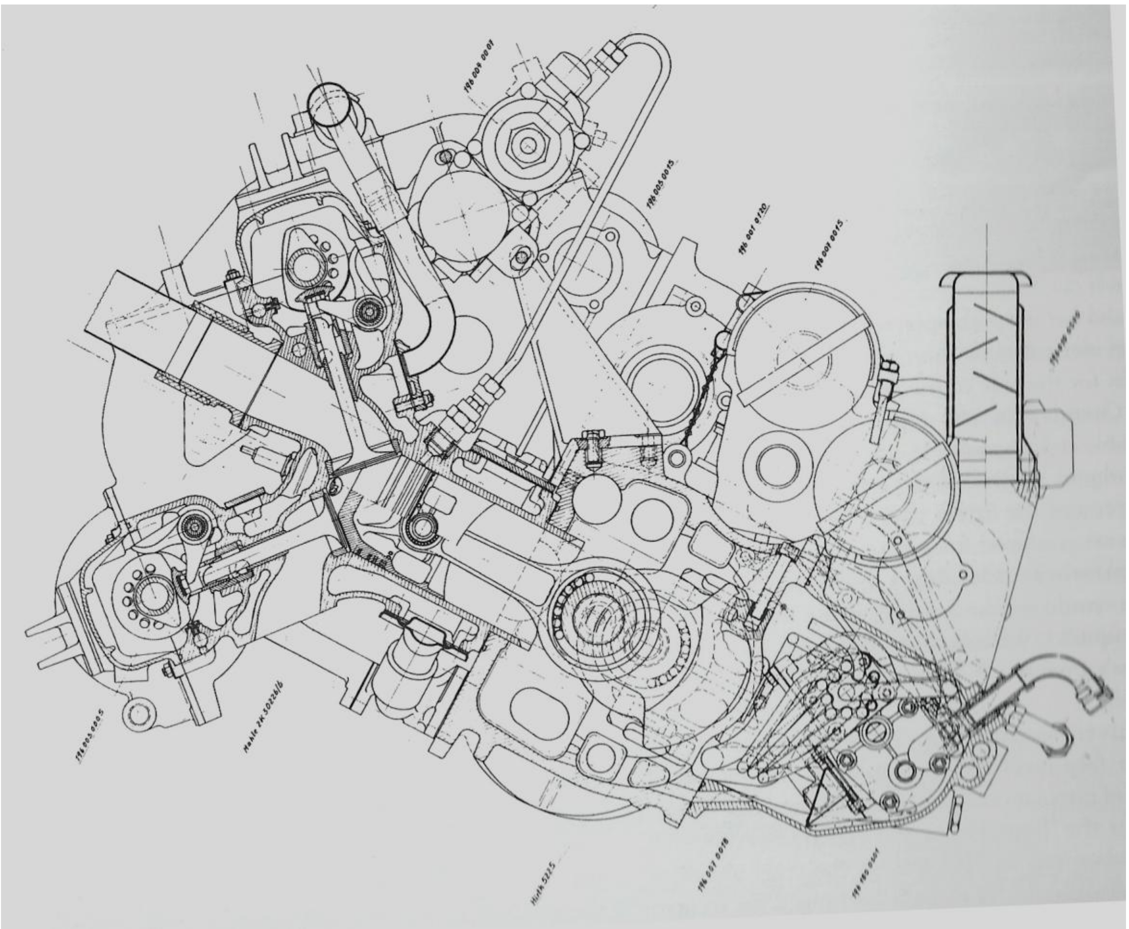
M196 Power

As well as the disappointment described over the need for dual crank dampers, “*Faces must have fallen somewhat*” (in Laurence Pomeroy’s words (32)) in early 1954 when the prototype M196 was first tested. For mid-1954 racing 257 PS was available (468) but the prototype may have been as much as 10% less than this – 254 PS – since 4 dual-choke Weber carburettors had to be used until the Bosch fuel injection system was available. The contemporaneous much-simpler and cheaper Maserati 250F *customer* engine with Weber carburettors was then giving a genuine 222 BHP (225 PS) (40).

However, it is significant that the M196 engine type *held* its power. The Sports-Racing 3 L version for the 300SLR of 1955 (IL8 3L B/S = 78/78 = 1) gave the same 300 PS *after* Moss won the Mille Miglia at a record 159 kph as *before* the race (986*).*

Fig. N53
1955 Mercedes-Benz M196.I (300SLR)
IL8 78/78 = 1 2,982 cc
This Sports-Racing derivative of the M196 2.5L Grand Prix engine had an Al-alloy casting in place of a steel fabrication for the combined cylinder head + block.

DASO 468



To return to home page: [Home Page]

Note 54

Combustion Chamber Shape and Efficiency

There are a number of examples of how Combustion Chamber Shape can affect Combustion Efficiency (EC).

(1). H.N. Charles in 1939 (294) described how, in the MG NE-type IL6 1.3L NA with bath-tub head having 2 vertical valves and R = 9.8, the combustion on 50/50 Petrol/Benzole was not complete until the piston crown opposite to the side-mounted sparking plug was raised "*so that...the gas....can see the flame*" as his foreman tester suggested.

(2). D. Munro of BSA (398) warned in 1949 of the chamber shape effect, writing:- "*....if the adoption of a high*" [compression] " *ratio involves the use of a piston with a very steep dome, there may be a loss of power due to the inefficient space thus formed*".

(3). Tony Rudd was advised similarly by Shell's Dr Harrow to cut off the top of the piston crown in the 1962 BRM V8 1.5L NA at the 14^0 angle of the plug so as to "*....show the charge to the spark*". Despite a drop of R from 12.2 to 11.2 this added 4% to Peak Power (40). The chamber was no longer "cut in two" by the high crown,

(4). Yoshio Nakamura (75) detailed how an increase in a basically-high-compression ratio could spoil the combustion chamber shape in a racing engine and produce a lower cylinder pressure (i.e. reducing the Combustion Efficiency more than the gain in Cycle Efficiency). His test results were obtained from a 25 cc 4-valve cylinder of the Honda 1963 RC113 50 cc twin, B/S = 33/29.2 = 1.14, VIA = 72^0 using high Octane Petrol. They showed that R = 9.3 with a smooth moderately-humped piston was more powerful than R = 10 when the latter needed 4 pockets in the crown to clear the valves with high-speed timing all open at exhaust TDC (note: (R x VIA) = 670^0 and 720^0 in these cases respectively). See Fig. N54.

Fig. N54
November 1963 Honda RC113
IL2 a/c 33/29.2 = 1.138 49.95 cc
Tested at 16,000 RPM:- MPS = 15.6 m/s.
DASO 75
Pictures approx. full size

	Piston A	Piston B
Compression Ratio	10	9.3
	Humped with deep valve-clearance pockets	Lower hump without pockets.

To return to home page: [Home Page]

Note 55

Maserati 250F Head Sealing

It is surprising that such a simple, one might say primitive, form of head seal did not cripple the Maserati 250F reliability, but it *could* cause trouble as described by Alf Francis (147) about the Moss-owned car.

At the 1955 International Trophy meeting in practice the engine lost water at a high rate despite 3 attempts to overcome the problem by removing and lapping the head. The trouble persisted and caused a seizure after overheating in the race. Only when the engine was taken back to the Maserati works was it found that No. 5 cylinder liner had dropped 2/1000'' so that escaping combustion gas forced coolant overboard. In theory the liner should have been fast in the block since the assembly process was to heat the Al-alloy block to 160^0 in an oil bath before pressing the liners into place against the retaining flange at 1/3rd depth (949). Presumably the block material had crept at that feature and allowed the liner to fall.

The problem must have affected other 250Fs although no details are known.

To return to home page: [Home Page]

Note 56

Maserati 250F engine price

Ref. (838) gives a mid-1954 price of 3.5M lire for a spare Maserati 250F engine as quoted to the Owen Organisation. This was about £2,000 at that date (£46,000 at mid-2013 money value).
This was half the factory-gate price for the whole car. For delivery in England this figure was then *doubled* by UK Import Duty and Purchase Tax (293)!

To return to home page: Home Page

8 February 2013.

Note 57

Squish in Maserati 250F?

Ref. (32) of 1963 by the distinguished authors Laurence Pomeroy and Stirling Moss illustrated by a sketch what was described as the shape of the 250F combustion chamber with squish plateaux fore- and aft of the 2 transverse valves. The drawing is inaccurate in that the 2 plugs, also fore-and-aft, are shown as widely angled apart – physically impossible – and external photos of the head and an engine section (949)(included in Eg.35) show that they were vertical in the longitudinal plane. The sketch artist clearly had not seen the parts he drew.

What *is* factual is a late 1957 (Pescara GP) photo of an inverted head of a customer (Centro Sud) 250F engine (795) which has *no* sign of squish plateaux. It is a simple hemispherical head, as in the transverse section quoted, although that drawing is probably a 1954 issue. See Fig. N57A.

In case the improved 1957 works cars included the squish feature of (32), the author has checked with Mr Neil Corner, owner of Serial No. 2528. He has confirmed that the original 1957 head still in his possession has normal hemispherical combustion chambers. He added that he had never come across a Maserati head as sketched (1081).

The 1957 Maserati 2.5L 60V12 engine cylinder head internals were shown in a photo (506) and, again. there was no sign of squish plateaux. See Fig. N57B

Actually there was no design problem in providing squish for a 2-valve head having wide VIA, if Maserati <u>had</u> chosen to do it, because Lampredi produced a head rather like the (32) sketch in the production FIAT 124S IL4 1.5L DOHC engine of 1966 (1082).

Fig. N57A
Maserati 250F cylinder head.
Some of the cylinders are damaged.
DASO 795

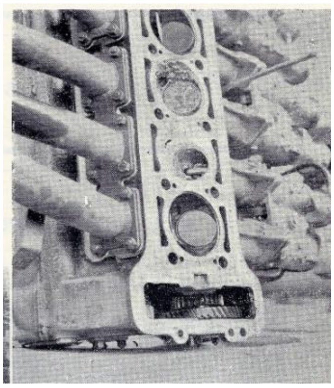
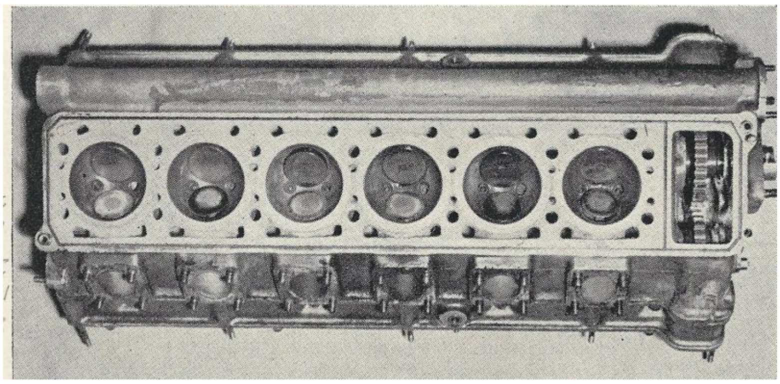


Fig. N57B
Maserati 60V12 cylinder head.
DASO 506



Note 58-2

Petrol development for commercial auto and military aero use

At the date of the Ricardo fuel tests in 1919 the search was already in progress for improvement in petrol anti-knock quality by crude oil source and by mixtures with benzole and alcohol. It continued with additives culminating in Tetra-Ethyl-Lead (TEL) (see Sub-Note A), in production by 1928, and better refining methods. These efforts raised petrol Octane Number (ON), the anti-knock scale defined by Edgar in 1927 (729) from a retrospective 40-to-50 pre-WW1 to 80 for the best commercial auto fuel and 100 for military aviation use at Stoichiometric Air/Fuel Ratio (SAFR) by 1937.

Rich mixture gain

It was then found in the UK that this 100ON fuel *in pressure-charged engines when run very rich* (about 60% rich, i.e. AFR = 9.2 instead of the usual Normally-Aspirated (NA) maximum-power setting of 20% rich, AFR = 12.2) would cool the engine* and so resist knock sufficiently to permit much higher charge inlet pressure to be used than would be expected from the Octane rating determined in the standard NA variable-compression calibration engine (the CFR 3¼'' bore unit). The increased pressure was sufficient to give 30% more power and the Performance Number (PN) scale was then introduced to rate the fuel as Grade 100/130PN, the two figures representing ON with a lean mixture and the extra power available on rich mixture (592,599).

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*Samuel Heron wrote in 1961 "*The use of rich fuel-air mixture as a means of preventing cylinder over-heating is still thought by many engineers to be due to the cooling effect of the evaporating fuel. There is reason to believe that this view is incorrect and that the reduction of cylinder temperature with rich mixture is due to reduced flame temperature*" (1057).

Improved rating use

The use of this improved rating was just in time to have a significant effect in the 1940 Battle of Britain and Grade 100/130 (which contained 5 cc TEL per Imperial gallon, 0.11% by volume) remained the main combat fuel used by the Allies in WW2.

Further development of aero fuel

Further development provided by 1944:-

(1). The US Grade 115/145PN for the air-cooled Wright R3350 (cubic inch displacement) 18 cylinder radial engines of the Boeing B29 Superfortress in very-long-range bombing operations over the Pacific (599);

(2). The UK Grade 100/150PN (100/130 with the extra additive of 2½% Mono-Methyl-Aniline) for fighter use in the European Theatre, initially to pursue the V1 flying bombs (598).

Power gains with higher fuel ratings

The advantage to be gained by increase in fuel knock resistance , permitting higher supercharge pressure, was illustrated in (598) from tests on the liquid-cooled Rolls-Royce Merlin 60^0V12 aero engine of 27 litres:-

Test Date	Fuel Grade	Engine Mark	Max. Sea-Level HP obtainable 150 Hour Type-tested	
Early 1939	87ON	XII	1,150	Datum
End 1942	100/130PN	66	1,750	x 1.52
Late 1944	100/150PN	RM17SM	2,200*	x1.91

*A 15 minute test was achieved at the end of 1944 with the RM17SM on 100/150PN + water injection at 2,600 HP (Datum x 2.26).

Improvements followed the general process of (1) fuel of higher knock-resistance; (2) supercharger improvement to make use of the better fuel; (3) mechanical development to make the engine reliable at the higher power.

As an indication of development over 26 years, it is interesting to compare the above figures with the rating of the 1918 US Normally-Aspirated Liberty aero engine, also of 27 litres, which was 400 HP on (retrospectively graded) 58ON fuel (901 discussion, Rod Banks' comment).

Triptane

Late in WW2 small quantities of a new fuel, Triptane, were produced rated at 140/200PN or, with 0.1% TEL added, 200/300PN. Had military piston aero engines continued in front-line use no doubt means would have been found to increase the supply for combat.

Post WW2 aero

However, post-WW2 the gas turbine, needing no fuel knock-resistance and therefore able to burn kerosene, terminated the military need for high PN aviation petrol and, after piston engines were also phased out of scheduled civil air transport, 100/130 remained the standard for private general aviation use. A halved-TEL version was introduced in recent years for environmental-protection reasons .

Post WW2 auto

Commercial auto fuel improvement continued to a peak as "5 Star" of 102 Research* ON in 1961. This was withdrawn from pumps in 1975 and work from 1986 was aimed at achieving 96RON without TEL, both moves for protection of the environment.

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*The post-WW2 "Research" test (for RON) is run at lower speed and lower inlet temperature conditions in the calibration engine compared to the "Motor" test (for MON) and gives 7 to 10 ON numbers higher (610). Previous tests for ON are understood to be equivalent to MON (714).

Sub-Note A

The very-high value of Tetra-Ethyl-Lead (TEL) as an anti-knock additive was discovered in the General Motors research laboratories fuel section under its chief Thomas Midgely (or Midgley – references differ) and his assistant Thomas Boyd in late 1921 (592). It is more effective per unit mass than any of 46 other tested chemicals listed in a 1938 source (594, Table 10), far more than most. Ref.(592) says 30,000 compounds were considered by GM! Compared with 2 other additives which were marketed later, benzole and ethyl alcohol (the UK petrol brands with these additives were "National Benzole" and "Cleveland Discol") TEL was x332 and x161 more effective, respectively. An addition of 2cc per Imperial gallon (2cc/Ig = 0.044% by volume) increased the ***zero*** Octane Number (ON) of Normal-Heptane to 40 linearly although the return decreased after that so that 5cc/Ig (0.11% by volume) achieved 55 ON (594 Fig. 104; these ON are retrospective to Edgar's 1927 scale).

To prevent objectionable deposits and corrosion in the cylinder it was found necessary to mix TEL with Ethylene Dibromide as a scavenger of the combustion products.

The 1st use of TEL in motor racing was at the 1923 Indianapolis 500 mile event (6).

Enviromental objections

After 50 or so years of widespread use, however, TEL in auto exhausts started to be blamed for low IQ in children living near busy roads. Although the connection was later disputed and an alternative explanation of the problem provided, the use of TEL began to be restricted from 1978. Shell developed unleaded racing fuels over 1988 – 1991 (535) and FIA rules required such fuels after 1992. TEL was banned altogether from UK auto standard petrol from 1 January 2000.

It is now fashionable to ignore completely the power/weight and fuel economy benefits which were provided by TEL in high-compression or high-supercharged engines (such as helping to win the 1940 Battle of Britain) and disparage Midgely's work (e.g. a newspaper article in 2000 headed "*The deadly Dr Midgely gets it wrong – again*" which added blame for the reported ozone-depletion effects of the refrigerator gas Freon to TEL (888)).

From the comfort and high degree of safety of a standard of living built on scientific and engineering advances it is quite usual nowadays to concentrate on the unforeseen (and unforeseeable) side effects of some of these advances - which have certainly *benefited* the lawyers – and to load all current research with every conceivable test against remotely-possible, if improbable, undesirable by-products. It has been said that if water were discovered today its unfortunate effects, in drowning, flood damage, ice and snow damage, iron rusting and timber rotting, would rule it out of use.

It only remains to be added that from 1979 up to 8% by volume of Methyl Tertiary-Butyl Ether (MTBE) was included in US gasoline to replace TEL for octane improvement, increased in 1992 to 15% to further oxygenate the fuel to reduce certain tailpipe emissions and meet 1990 legislation (creating “Reformulated Gasoline, RFG), that MTBE traces were then found to be polluting drinking water, that legal actions ensued claiming it to be carcinogenic and that California (always wanting to be forward in environmental issues) in March 1999 ordered it to be phased out of state fuel by end 2002 (later extended to end 2003)(893).Ethanol at 75% may replace it.

[Written originally in early 2003. The author has not attempted to follow later twists in this saga, which led to 6% of ethanol in state gasoline and even some use of 85% (!), only to note that in early 2012 California is causing complaints that it does not want US-grown corn-based ethanol but prefers sugar-cane based imports from Brazil, allegedly better for the environment – but also wants to get rid of any transport system which emits CO_2, thereby again upsetting the ethanol industry which expanded to serve their earlier enthusiasm.]

Note 59

Development of the Ferrari F2 into F1, 1957

On this subject, after consulting 6 authors (Jenkinson (502, 1958); Tanner (8, 1959); Casucci (22, 1980); Laban (390, 1990); Ludvigsen (711, 2001); and Acerbi (1077, 2004)) who sometimes agree, sometimes contradict each other, sometimes quote Bore and Stroke which do not correspond to the quoted capacity and none of whom tell the whole story (pity the historian!), the following appears to be the best narrative of this development.

Date	Type	65V6, DOHC Bore Bmm	Stroke Smm	Swept volume Vcc
April 1957	156F2	70	64.5	1,489

Raced at Naples (see Fig. N59A) and Rheims, then the 2 engines made were enlarged.

22September 1957 @ Modena: 2 engines.

	Type	Bore	Stroke	Swept volume
	?	≈78.2	64.5	1,859

Jenkinson, very much the "man on the spot", wrote: 1st engine "nearly 1,860", 2nd "slightly smaller", crank the same as 156F2. Presumably enlarging the bore for "nearly 1,860" was a trifle risky hence the "slightly smaller" bore for the 2nd engine and the need for a Larger Cylinder Block for a further increase in capacity:-

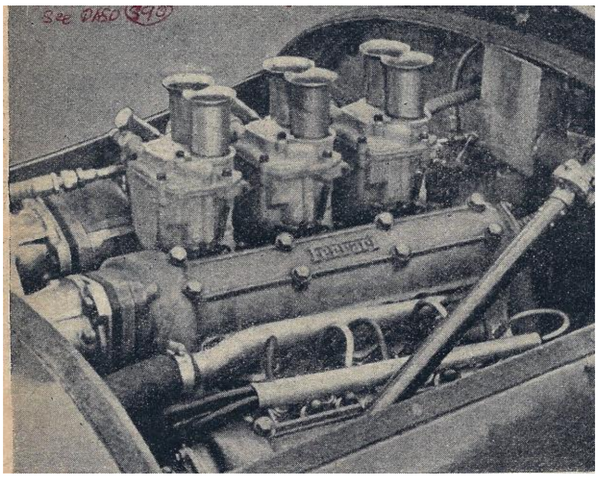
27 October !957 @ Casablanca

	Type	Bore	Stroke	Swept volume	
1st engine	?	85	64.5	2,196	Original crank.
2nd engine	246F1	85	71	2,417	New crank.

Why Ferrari in 1957 did not go immediately to 86mm Bore for 2,475cc, as was done for 1959 (type 256) is a mystery.

Three prototype Sports-Racing engines were produced in 1958 from these 65V6 bases (8);- respectively types 206S: 77 x 71 = 1,984cc; 226S: 81 x 71 = 2,195cc; 296S: 85 x 87 = 2,962cc.

Fig. N59A
April 1957 Ferrari 156F2
DASO *Motor* May 1957



Note 60

The 1952 500 cc Norton engine in Vandervell's Cooper

While design and construction of the Vanwall engine was in hand in 1952/53, Tony Vandervell thought it would be useful to have experience of the "base" engine, the 1952 works Norton, B/S = 85.93/86 = 0.999 and 498.7 cc, by racing it in a Cooper Mk 7 chassis.

This car was driven by Alan Brown, a very experienced "Half Litre" exponent, in 2 races in 1953. He found very quickly that the limited RPM range with only a 4-speed gearbox made it impossible to produce competitive lap speeds (68).

Rival cars also had Norton 500 cc engines but these were production "Manx" 79.62/100 = 0.796 units with somewhat less peaky power curves. Moreover they were certainly running on alcohol-base fuel where the '52 ex-works engine may have still been on motorcycle racing regulation 80 Octane petrol.

Note 61

Coventry Climax type FPE versus potential rival Grand Prix engines

In 1954 the leading characteristics of 2.5L Grand Prix engines were as tabled below.

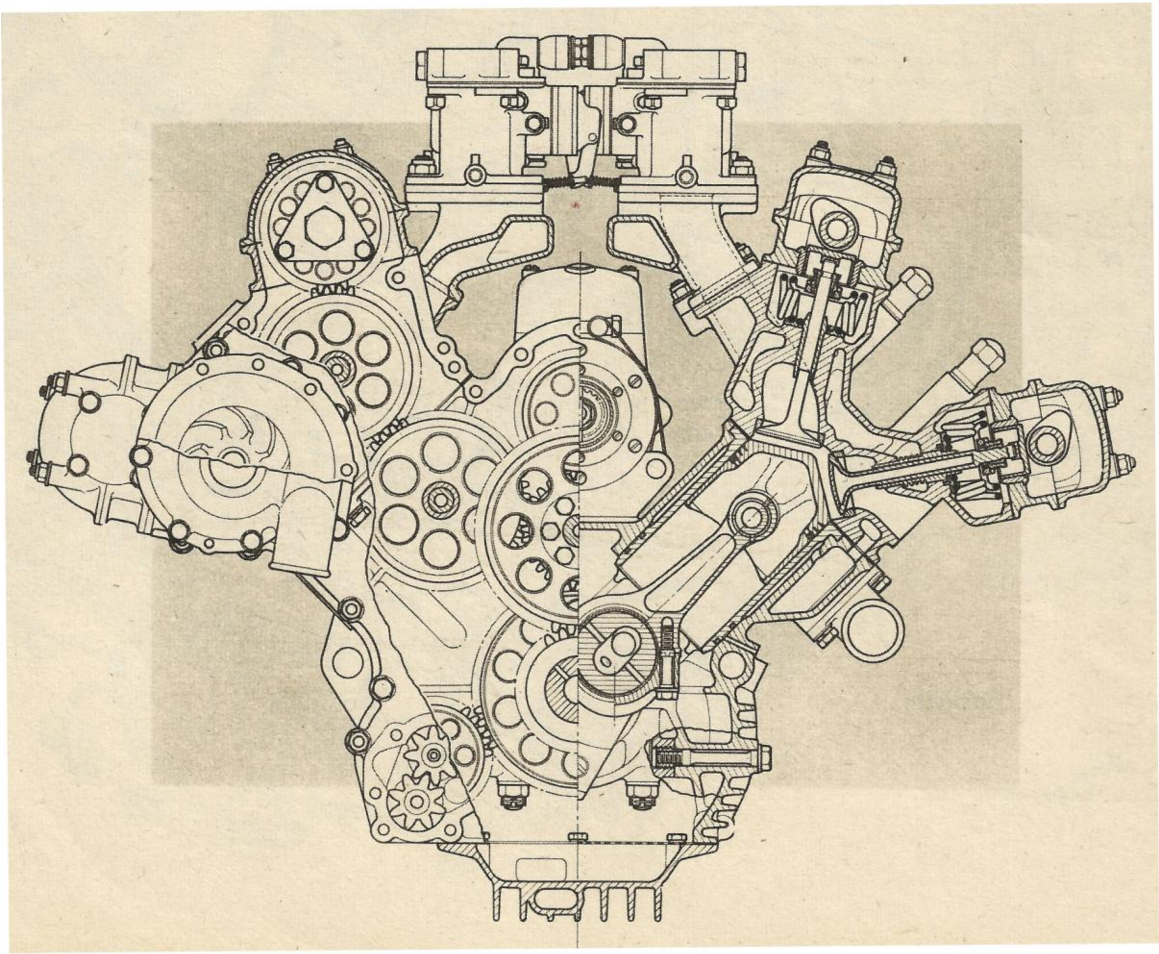
Make	Mercedes	Maserati	Ferrari	Lancia	Climax
Type	M196	250F1	554 (?)	D50	FPE
Data sources	468	147, 158	8, 22, 711	800, 1089	33, 131B
Configuration	IL8	IL6	IL4	90V8	90V8*
B/S mm/mm	76/68.8	84/75	100/79.5	73.6/73.1	76.2/67.945**
	= 1.105	1.12	1.258	1.007	1.121
					**3''/2.675''
R	12.5	12.5	12	10.5	12.3
Methanol in fuel	25%	50%	Yes, ?%	Yes, ?%	65%
PP HP	253 (n1)	214 (n2)	240	260 (n4)	258 (n6)
@ NP RPM	8,250	7,000	7,500	8,000	8,250
TP lb.ft.	183	180 (n2)	?	?	190 (n6)
@ NT RPM	6.300	6,000	?	?	6,000
$\frac{NP - NT}{NP}$	23.6%	14.3%	?	?	27.3%
W kg	205	178	160	170	154
PP/W HP/kg	1.23	1.20	1.5	1.53	1.67**
					**(36% higher than the M196).
Major races won	4	2	2 (n3)	None (n5)	Not raced (n7)

Notes
(n1). 257 PS
(n2).UK test of Stirling Moss' engine after full works overhaul (147). Hassan quoted 230 HP as best used by Fangio in early 1954 (575).
(n3). ! win in a type 625 chassis with a 625 crankcase.
(n4). Other sources quote 250 HP.
(n5). Fastest lap in the last race of the season but this may have been with a light fuel load in a car not expected to last its 1st race.
(n6). Table 4 and Fig. 24 in ref. (33). Text of (33) quotes 264 HP @ 7.900 RPM.
(n7). Source (131B) stated test in 1954.

*A section of the Coventry Climax FPE is given on P.2.

1953 Coventry Climax FPE
90V8 3''/2.675'' (76.2mm/67.945) = 1.121 2,479 cc
This is the original design with hairpin valve springs (HVRS).
The inlet tracts are angled to give radial swirl.
The exhaust valves are hollow in both stem and head for an internal coolant.
The exhaust valve guides have a portion in direct contact with water.
The con.-rod is split at an angle to permit withdrawal upwards.
There are extra transverse bolts retaining the main bearing caps.

Data Source Autocar 7 August 1953.



Contrast the type FPF in 2.5L form shown on Fig.39A.
This IL4 engine in 1960 had PP/W of 1.82 HP/kg on AvGas 100/130 fuel, 8½% higher than the 90V8 FPE on 65% methanol.

The performance of the family of Coventry Climax DOHC racing engines is given in Note 20.

Note 62

Effect of CRL/S on performance

In a separate investigation of engine data the author found that there was an increase in performance as CRL/S rose.

Hassan and Mundy chose for the Coventry Climax FPE a ratio of CRL/S = 5’’/2.675’’ = 1.87. Then this 5’’ con.-rod was retained for convenience and cost in developing the FPF series, although altered from an angled joint to straight, until the final 2.5L stretch. It was then thought worthwhile to add 0.1’’ (2%) to the length, presumably the most that could be obtained with the forgings from the original dies. Therefore the 2.5L unit had CRL/S = 5.1’’/3.54’’ = 1.44.

In contradiction to the above statement, Tony Rudd found that he obtained a 6% power gain from his IL4 1 Litre Formula 2 engine when CRL/S was *reduced* 11%, from 4.625’’/2.425’’ = 1.91 to 4.125’’/2.425’’ = 1.70 (40). He believed that there was a rise in Combustion Efficiency from the increased piston acceleration away from Top Dead Centre which was more than offsetting the increased piston friction.

With a free choice for the 1967 DFV Keith duckworth used 5.23’’/2.55’’ = 2.05 and that engine certainly was not short of performance!

There *has* been a move to higher values of CRL/S with time, egs.:-

1992 Honda RA122E/B	[not Coy]	2.32	(69);
2000 Ferrari 049		2.68	(987);
2009 Toyota RVX-09H	[not Coy]	2.72	(1091).

Note 63

1949 Cooper versus Ferrari in Formula 2
(and the development over 10 years)

Ferrari V12 2L engines dominated early Formula 2. In 1949 it seemed possible to some commentators that a Cooper 500 cc chassis fitted with an existing JAP 55V2 1,000 cc aircooled engine, mid-mounted, could beat the front-engined "traditional-chassis" Ferrari , or other similar 2L cars, by virtue of a 10% higher Power/Laden Weight ratio. This did not turn out to be the case in races of any great distance, even on low-speed circuits (egs. Isle of Man, Lake Garda). This was partly due to unreliability. Certainly the Cooper could not defeat the Ferrari on a high-speed circuit (Rheims). True, the Cooper effort was not a massive one relative to the works Ferrari entries in some events. In 1963 Laurence Pomeroy pointed out in ref. (33) (in a simplified analysis) that the influence of drag on the acceleration curve, even at medium speeds, had been overlooked, the Ferrari having around 20% more power in relation to frontal area.

A redesigned JAP 55V2 1,100 cc engine for 1950 did not change the situation, Ferrari having moved on also.

Cooper then turned in 1952 to the more powerful and reliable watercooled Bristol IL6 2L engine, front-mounted – by which time Ferrari had moved even further ahead with the Type 500!

However, via 1,100 cc Sports-Racing and then 1.5L Formula 2 cars, Cooper came to the Grand Prix line in 1959 with mid-engined vehicles substantially superior to Ferrari except on high-speed circuits. In 1960 that performance shortage was rectified with a lower car.

The development of Cooper cars is tabled below.

Cooper chassis

Date	Car	Wheelbase Mm	Front Track mm	Engine Mounting
1948	500 cc	2159	1245	Mid
1949	1,000 cc	"	"	"
1952	2 Litre	2286	1270	Front
1955	1,100 cc Sports-Racing	2261	1156	Mid
1957	1.5 Litre Formula 2	2311	1156	"
1959	2.5 litre Grand Prix	"	1181	"
1960	"	"	"	"
		+ 7% of 1948	-5% of 1948	

The Power/Weight advantage of increasing installed power is shown by these figures, where a 240 HP engine was installed successfully on a wheel plan area only 1.5% larger than that of the 38 HP car of 12 years earlier.

The cars all ran on 15'' diameter wheels with tyre sections which were increased gradually from 4.00'' all round to 5.00'' front and 6.50'' rear.

An illustration of the new 1950 JAP 1,000 cc engine, representing the 1,100 cc, is given on P.2.

1950 JAP (J.A.Prestwich) 1,000 cc

55V2 80/99 = 0.808 995.3 cc

Al-alloy cylinder barrels with cast-iron liners, Al-alloy heads, Mg-alloy (“Elektron”) crankcase.

The Con.-Rod assembly is “fork-and-blade”.

Representing

1950 84/99 = 0.848 1,097.3 cc

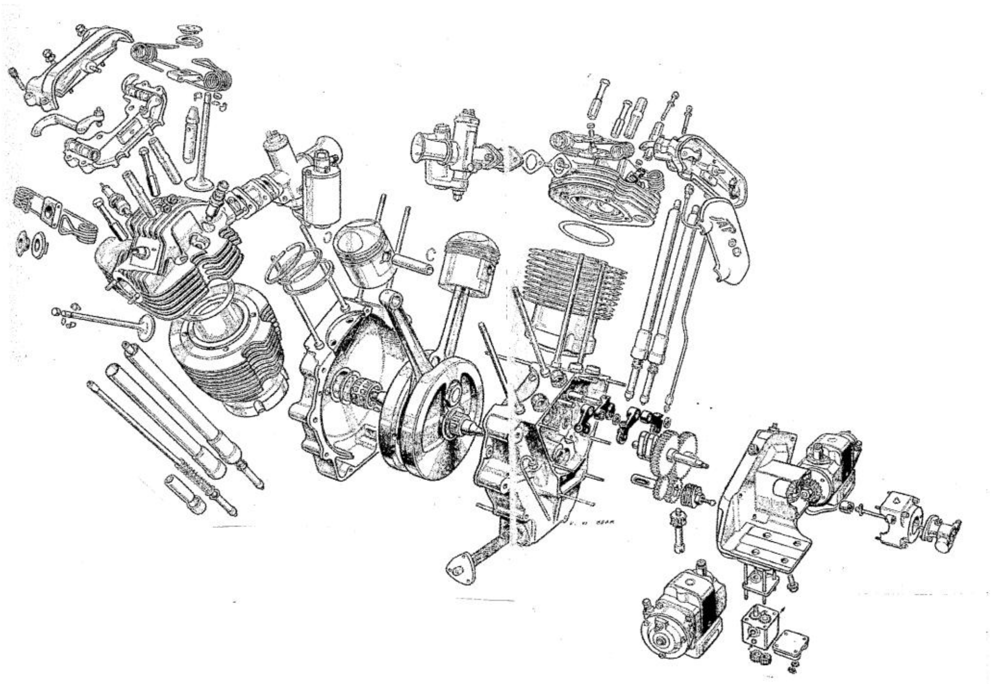
R = 14; Methanol fuel.

Peak Power (PP) = 95 BHP @ 6,000 RPM.

Weight (W) = 56.7 kg.

PP/W = 1.67 BHP/kg.

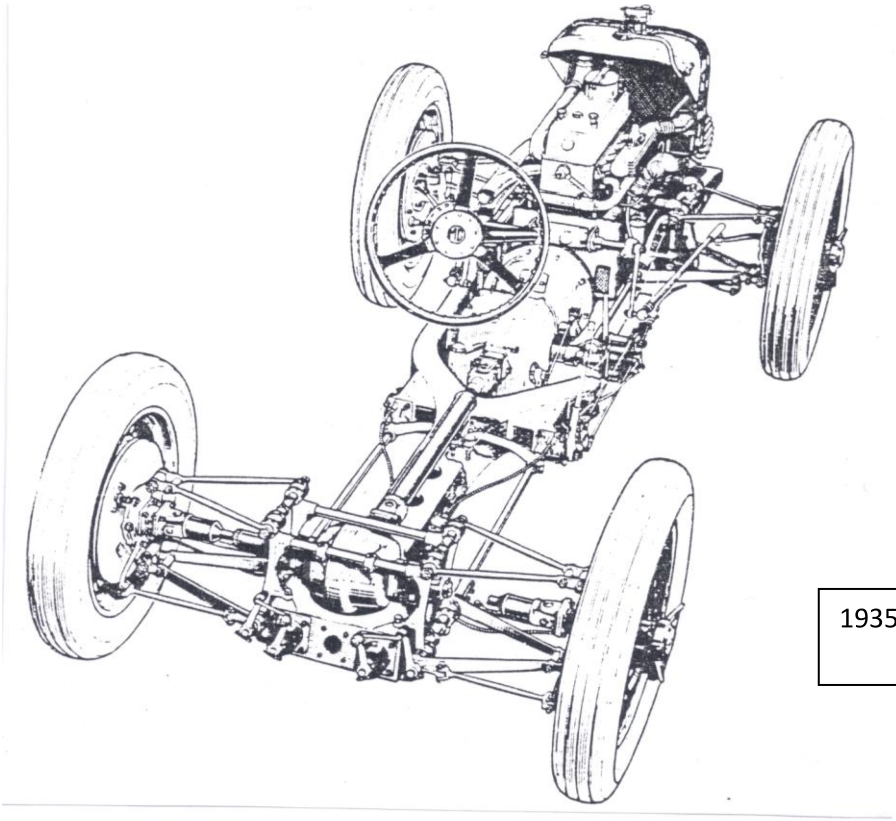
DASO 73



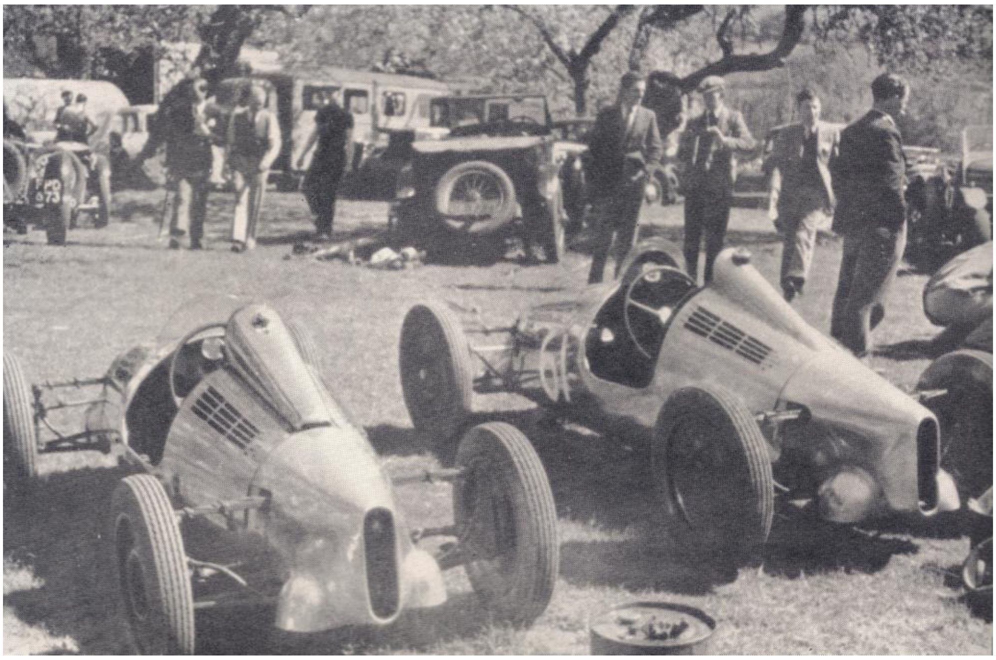
Note 64

Grand Prix race distances, 1957 – 1960

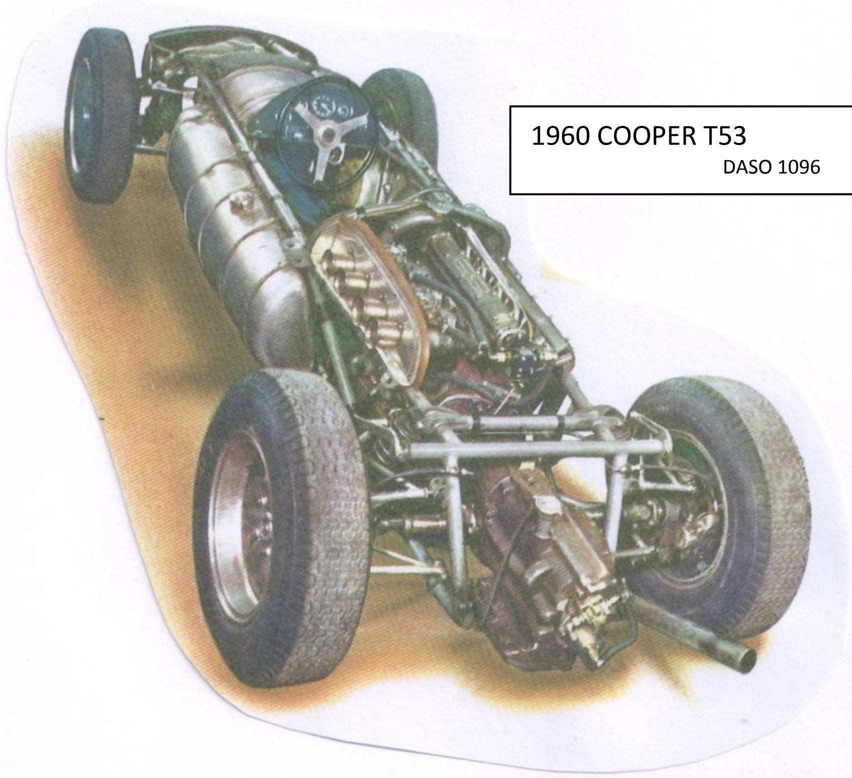
Year	Fuel	Championship Races – No.	Average Length km	Range km	Average
1957	Free	7	446	330 – 504	Datum
1958	AvGas 100/130	10	357	313 – 415	80.0%
1959	”	8	376	314 – 498	84.2%
1960	”	9	392	313 – 508	87.9%



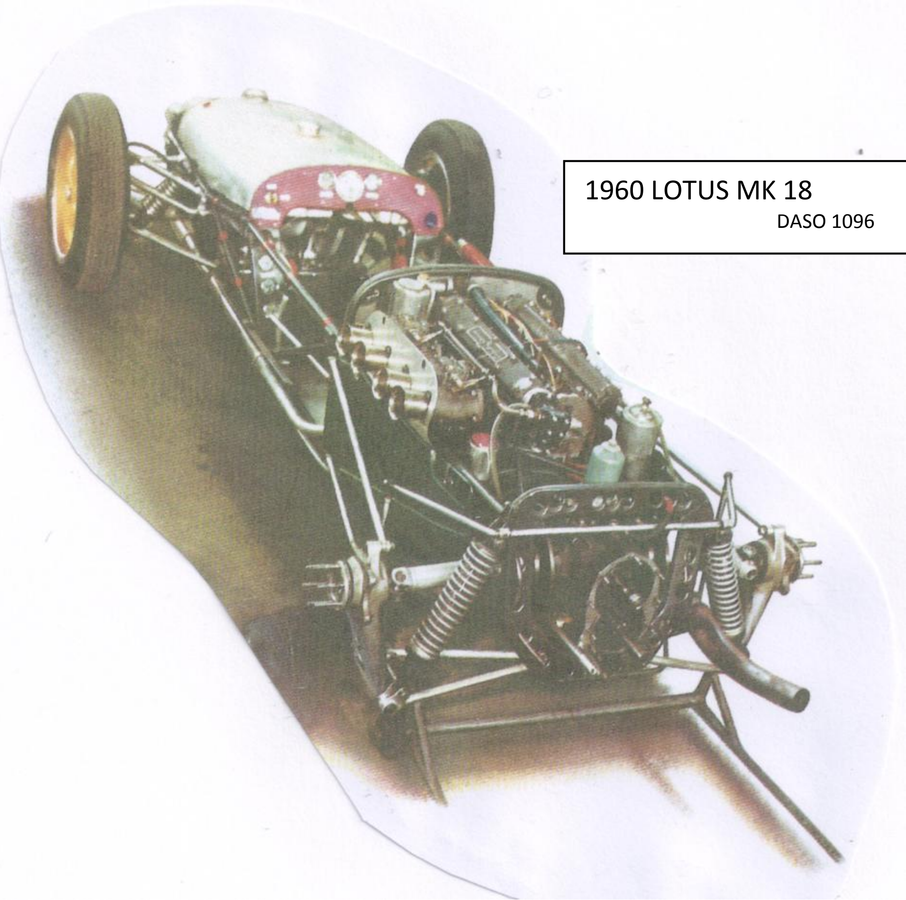
1935 MG R-type
DASO 806



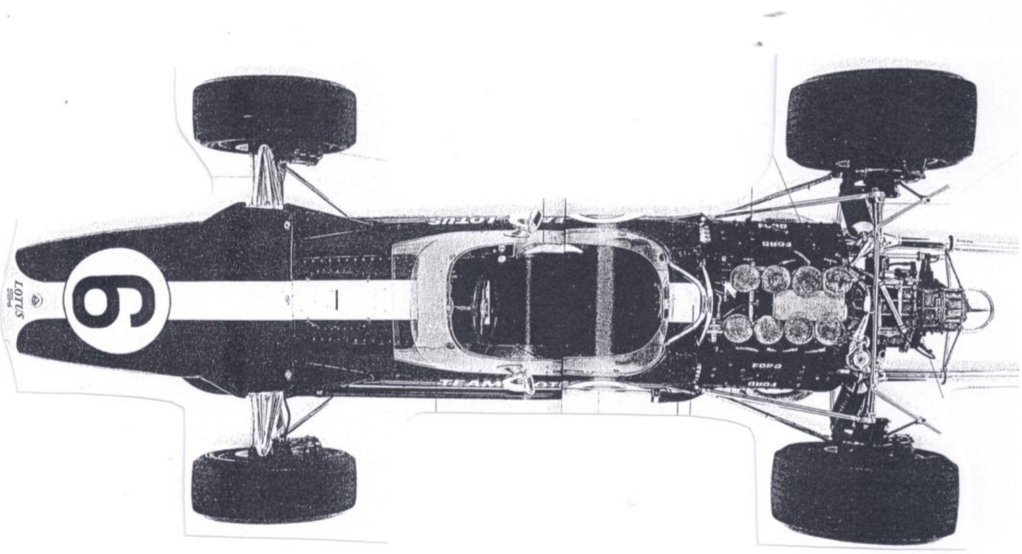
1946 The first two Cooper 500s
DASO 483



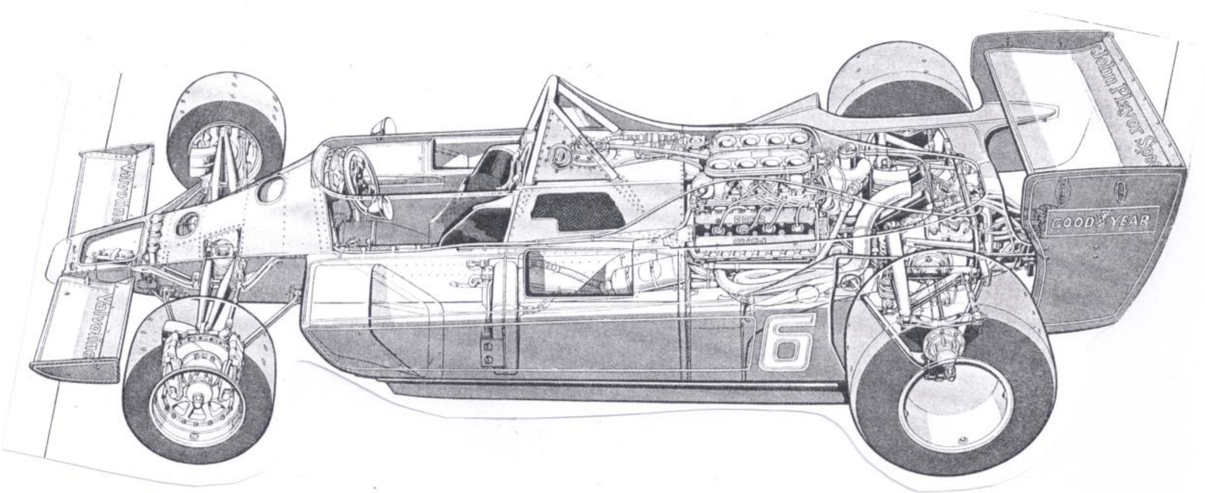
1960 COOPER T53
DASO 1096



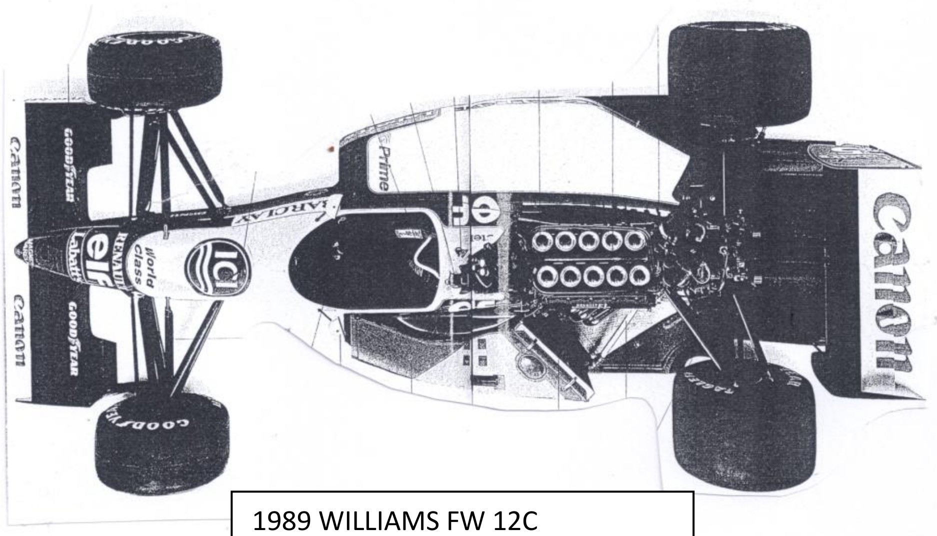
1960 LOTUS MK 18
DASO 1096



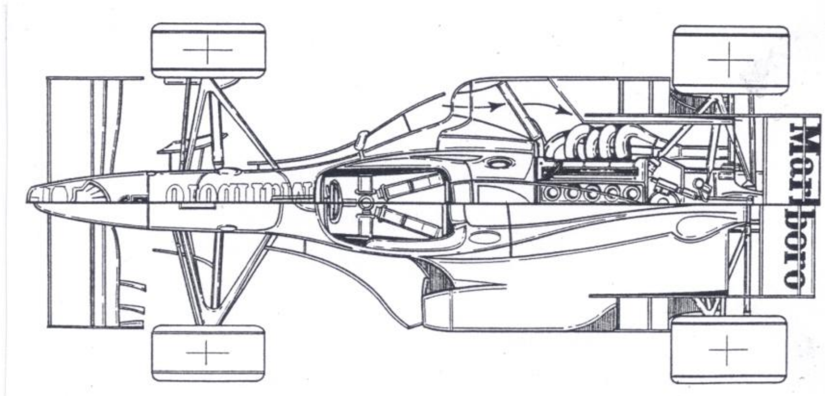
1967 LOTUS MK 49
DASO 964 p.96



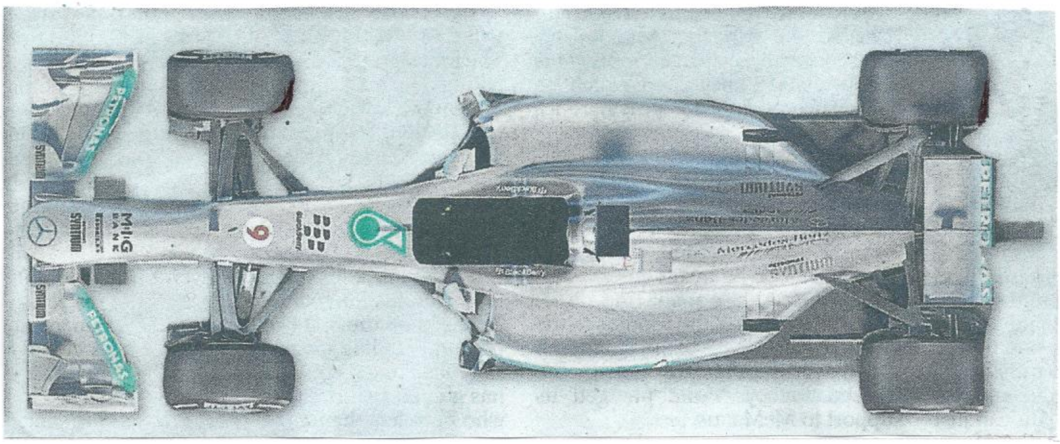
1978 LOTUS MK 79
DASO 961



1989 WILLIAMS FW 12C
DASO 964 p. 132



1996 FERRARI F 310 Top ½ plan
1997 FERRARI F 310B Bottom ½ plan
DASO 707



2013 MERCEDES-BENZ
AMG WO4
D/Tel 16 March 2013

Note 66

The 'Standard' Grand Prix suspension

(The contributions of Hubert Charles, Charles and John Cooper, and Colin Chapman)

Insofar as a modem Grand Prix car, with vast aero downforce, has any suspension – *"grovelling around a few millimetres above the road"**, as Keith Duckworth once remarked acidly, and then being rammed over kerbs with dampers solid in shock to prevent the all important floor from being torn off the car - its basic layout goes back to the R-type MG of1935.

The R-type MG

At that date Hubert Charles designed a racing chassis suspension which differed completely, not only from the rigid-axled layouts of all successful pre-1934 racing cars and previous MGs, but also from the 'new order' introduced to GPs in 1934 by Mercedes-Benz and Auto-Union. These latter cars had independent front and rear suspension (IFS and IRS) with an inclined roll axis, low front/high rear, giving an oversteer characteristic very embarrassing with high power/weight ratio. Charles' solution was double transverse links at each corner. With links initially parallel to the road, the roll axis was at ground level. Front and rear suspensions were connected by a very stiff box-backbone chassis. A somewhat high centre-of-gravity (CG) height/track ratio meant that the R-type rolled considerably on corners (a photo shows an outer front wheel at 10^0 positive camber on a Brooklands 'road' circuit hairpin (810)), to the unease of drivers used to near-flat cornering and, admittedly, to the reduction of cornering force from the positively-cambered wheels. There was also teething trouble with the dampers. However, wheels which leaned but which stayed on the road over the bumps of pre- WW2 circuits, because of the reduction of unsprung weight, were greatly preferable to rigid-axle wheels hopping into the air (139)! It is certain that the drivers would have got used to the roll in time** and would then have appreciated the above-mentioned and other advantages of IFS and IRS <u>plus</u> the stable steering. Unfortunately, Lord Nuffield had lost his limited enthusiasm for motor-racing after the accidental death of an MG mechanic in 1934, sold his personally owned MG Co to his parent group in 1935 and then permitted Leonard Lord (Group Managing Director) to close the racing department abruptly only two months after the R-type debut (806). The car, therefore, had no chance to show what it could do after the skilful development which Charles could have carried out. His plans for an 'active' Mk 2 R-type, in which a gyroscope would have commanded hydraulic cylinders to move the longitudinal torsion-bar spring-stops and jack the chassis against roll (806) might have been too elaborate for the '30s but transverse roll-stiffening torsion bars at each end were not beyond his brilliant mind. Knowledge was just coming across the Atlantic of Maurice Olley's experiments in the USA where, in 1933, an experimental Cadillac had been fitted with a front bar to alter the steering characteristic (807). Of course, such bars *do* convert IFS and IRS into semi-independent systems and in proportion to their stiffness do bring back some of the rigid-axle disadvantages but this point will not be laboured here!

The Cooper 500

The R-type was a very promising idea, which did not progress through no fault of its own. A racing mechanic of its era, Charles Cooper, must have been familiar with it. It would be stretching the connection too far to suggest that it was in his mind when, post-WW2, he and his son, John, built their first 500cc car. The availability of several IFS units from scrapped pre-WW2 Fiat 500 'Topolino' road cars was no doubt the deciding factor, this IFS having double transverse links for each wheel made up of a lower wishbone and an upper centrally-mounted transverse leaf spring - but the stroke of genius was to use these assemblies <u>at each end</u>!

* Admittedly regulation changes since 1993 have forced cars a little further off the road!
** As Michael Hawthorn did in the 1952 Cooper-Bristol, qv.

This layout, gradually refined in springing and links location by the Coopers with Owen Maddock (their only designer) and later with the input of Jack Brabham when he became their No 1 driver, was the unique 'trademark' of all Cooper cars for the next two decades. It became the 'standard' way to suspend a Grand Prix car from 1960. The mid-engined layout also pioneered post-WW2 by the Coopers (except for the 1952-1953 F2 cars and a few sports cars) kept the CG height/track ratio fairly low and so the roll angle *could* be much less than the MG R-type. However, a photo shows a 1952 front-engined Cooper-Bristol (high CG) cornering at speed in the hands of Hawthorn with 7° of positive camber on the outer wheels (see Fig. N66A on P.4) and another of a 1952 500cc car (lower CG) driven by Eric Brandon with 8° (Fig. N66B). These figures can be compared with a 1951 Alfa Romeo 159 in a full-blooded power-slide by Fangio in the British GP having 4° positive camber on the outer front wheel and 0° on the outer rear - this particular car retaining swing-rear-axle set up statically with negative camber (Fig. N66C)

Reducing the roll angle

Steps, therefore, were taken gradually by the Coopers to reduce this roll. The 1953 500cc chassis picked up the post-WW2 Fiat 600 *semi-IFS* improvement with the transverse leaf spring held between two spaced-apart roller mountings, plus a central lateral location with vertical freedom, so as to give a higher spring rate for roll than in pitch; the 1954 500cc Mark 8 applied this scheme at both ends*. The system was continued on larger-engined cars until their higher speeds called for IFS by double wishbones and coil springs with a torsional roll-stiffening bar (hereafter 'anti-roll' bar) in early 1958 (type 45). Shortly afterwards double wishbones were adopted at the rear, although still with the roll-stiffened transverse leaf spring. This car, with which Jack Brabham won his 1st Championship, cornered at speed with roughly half the outer wheel angle - 4° - of the earlier types mentioned, although Stirling Moss in a 1959 car 'in extremis' while trying in an early 1960 race to catch a new Lotus 18 was photographed at 6° with the inside front wheel off the ground (Fig. N66D) - but a wishbone broke after he took the lead (1043).

The Lotus 18

The Ll8 was Colin Chapman's Cooper-inspired first mid-engined chassis with double wishbones, coil springs and anti-roll bars at each corner. The Lotus outer front wheel angle in the photograph mentioned was 3°. The Coopers, having seen earlier L18 performances in 1960, had already designed the new lower T53, in which the IRS was made basically the same as the 1958/1959 IFS. Brabham won a 2nd Championship with this. Illustrations on P.5 show the Cooper improvements from 1957 to 1961.

Thereafter, while there might be more widely-spaced wishbone elements to reduce loads and so permit lighter parts - eg the L18 rear suspension, which also used fixed-length jointed half shafts to double as wishbone elements (812)** - the 'standard' suspension pattern for GP cars had been established.

Given the basic 'double transverse link at each corner' formula, detailed variations could be used: relative lengths of upper and lower links; front/rear anti-roll bar relative stiffness rates; front/rear weight distribution; tyre sizes and pressures; damper settings; cambers. Some would be fixed at the design stage, some could be varied at the circuits to produce a steering characteristic which (at least attempted!) to satisfy a driver's wishes.

The desirable steering characteristic

Laurence Pomeroy, writing in mid-1963 (804) gave Colin Chapman the credit for being the first racing chassis designer with his L18 *"consciously ... to deduce that the optimum solution*

---

* Bob Gerard applied torsional roll-stiffening bars to each end of his front-engined Cooper-Bristol at about the same date.

** The LI8 rear followed Frank Nichol's 1958 Elva Mk 4 sports car layout, also used on Eric Broadley's 1958 Lola sports car. It appears that the use of a fixed-length half shaft as a suspension element had been patented by Georges Roesch in 1934 (814)!

was tameable over-steer. which the super-driver could induce at will by braking or accelerating". Actually Jack Brabham, the No 1 Cooper driver and World Champion in 1959 and 1960, was already on record as preferring mild under-steer provided that there was sufficient power to create over-steer on demand (811).

Later in 1963 Pomeroy acknowledged (32) that Rudolf Uhlenhaut had modified the Mercedes-Benz W196 before the 1955 season to have neutral-to-final over-steer instead of the strong 1954 under-steer. He also accepted that the Ferrari and Maserati designers of the '50s achieved the same final over-steer characteristic - but felt, perhaps, that in their cases (to quote from the Foreword of this review) it was *"a mixture of Inspiration ... and Experience"* and that Chapman had inserted '*Calculation*' into this mixture at the design stage.

That the desired end result could be achieved in more ways than one by the details in producing the 'now standard' GP chassis was illustrated by Stirling Moss in describing (32) (815 provides the details) how he was able to lap Warwick Farm circuit (in Australia) in February 1962 1% faster in a 'lowline' 1960-type T53 Cooper than a 1961 Lotus 21. The former had a 2.7L FPF, however, and the latter a 2.5L, and that must have been worth a couple of percent around the twisty course. Moss conceded that the Lotus would probably have been faster, driven with complete attention, whereas the Cooper was more forgiving and could be 'thrown' around*.

What was being realised consciously at this date in racing car design was the application of the 'Natural Rule of Thumb' well-known in aeronautical circles, whose particular derivative for vehicles is:-

$$\frac{\text{Stability x Control}}{\text{Complexity}} = \text{Constant**}$$

To visit 'Natural Rule of Thumb' click here:-
grandprixengines.co.uk/Natural_Rule.pdf

Technical revolution from minimal resources

It remains only to be recorded that (what has been entitled) 'The Cooper Revolution', of a mid-engined car with a winning mixture of stability and control, was produced by father and son and Owen Maddock from a tiny organisation with no greater resources than a small repair garage and which bought its own engines. Of course, as they and their drivers worked their way to the top, they did benefit from the free development strengths of Coventry Climax engines and Dunlop tyres, plus a subsidy from Esso to pay Brabham's salary (813).

Note 66 Illustrations.pdf

This linked companion section provides illustrations as follows:-
1935 MG R-type; 1946 The first two COOPER 500s; 1960 COOPER T53; 1960 LOTUS MK18; 1967 LOTUS MK49; 1978 LOTUS MK79; 1989 WILLIAMS FW 12C;
1996 FERRARI F 310; 1997 FERRARI F 310B; 2013 MERCEDES-BENZ AMG W04.

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* This was the last occasion on which a super-driver could compare two state-of-the-art cars from different makers, because both belonged to the last of the grand independent team owners, Rob Walker.

** This had been seen in the contrasting fighter aircraft pairs of 1917 and 1940: in WWl SE5A (stable, 'under-steer', a good gun platform) v. Camel (unstable, 'over-steer', easily manoeuvred in a dogfight - by pilots with exceptional reactions, ie 'super drivers'); in WW2 Hurricane (,under-steerer') v. Spitfire ('over-steerer').

Fig. N66A
1952 Cooper-Bristol T20
Michael Hawthorn at Goodwood, Easter Monday, winning the F2 race
DASO 483-2 *The Racing Coopers.* A. Owen. Cassell. 2nd Ed. 1959.

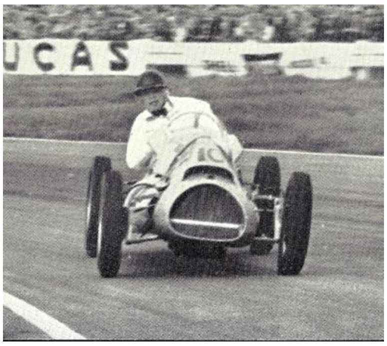


Fig. N66B
1952 Cooper-Norton 500 cc MkVI
Eric Brandon at Goodwood.

DASO 808

Fig. N66C
1951 Alfa Romeo 159
Juan Fangio at Silverstone.
DASO 809

Fig. N66D
1959 Cooper T51 versus 1960 Lotus Mk 18
Innes Ireland leading Stirling Moss in the 1960 International Trophy at Silverstone.
DASO 1043

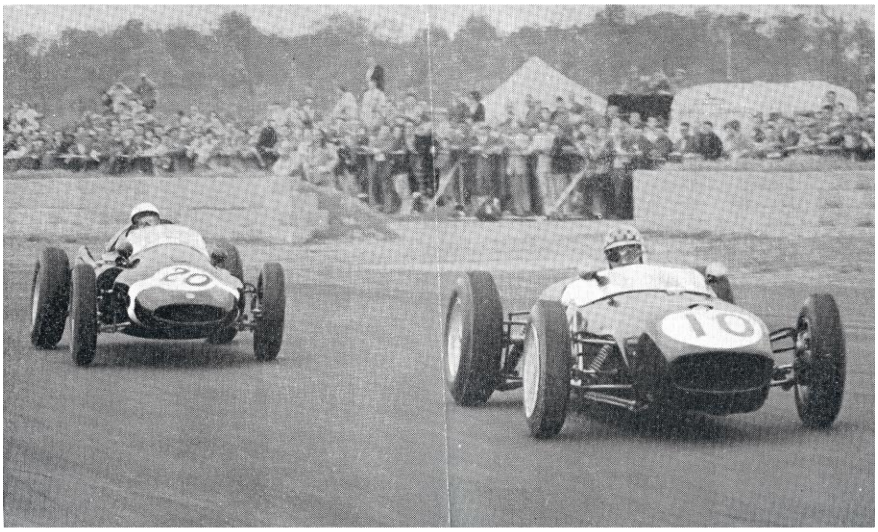


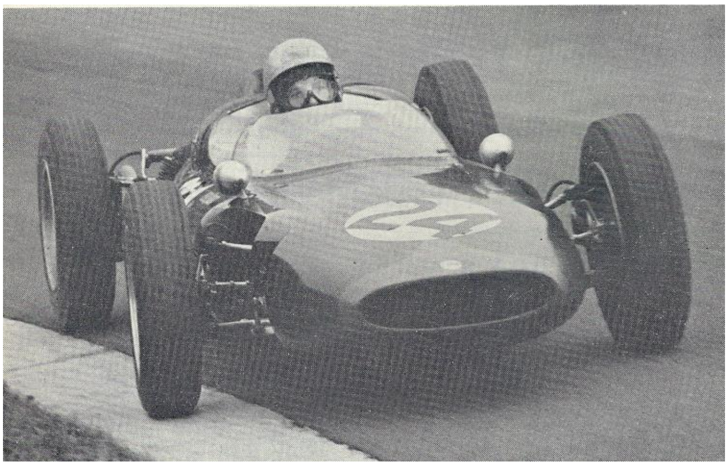
Fig. N66E
1957 Cooper Climax T42 Formula 2
Coventry Climax FPF 1½L engine.

The photograph below, showing Jack Brabham on Woodcote corner at Goodwood, was published on the front cover of the magazine *Motor Racing* for November 1957 (DASO 506).
Brabham wrote in ref. (811), published in 1960, "*I could see that the Cooper suspension needed a lot of attention.* [It] *showed that the car had much too much body roll, the wheels.... were leaning badly......When we went over to the double wishbone car in 1958* [these photographs were] *invaluable in helping to rectify the Cooper suspension*".

The car is in an oversteering slide with the driver applying some opposite lock.

Fig. N66F
1961 Cooper Climax T53 Formula 1
Coventry Climax FPF Mk2 1½L engine.

This photograph shows John Surtees in a Yeoman Credit-owned T53 with the FPF MK2 engine to the new formula in the Belgian GP on the climbing RH corner just beyond Eau Rouge, i.e. at very high speed. The suspension is compressed by the inertia load.
The improvement in the car's cornering with the "low-line" T53 chassis is clear
DASO *Motor Racing* August 1961



The car is in an understeering 4-wheel drift.

Surtees finished 5th, the first car after four Ferrari 156s which had up to 30 HP more power.

Note 66B
Two pioneers of the "Standard" Grand Prix suspension system

In 1948 – 1950, while Cooper were serendipitously using suspension by double transverse links at each wheel which they then developed eventually into the "Standard" Grand Prix car suspension system, two English-built racing cars used this layout as a matter of design choice.

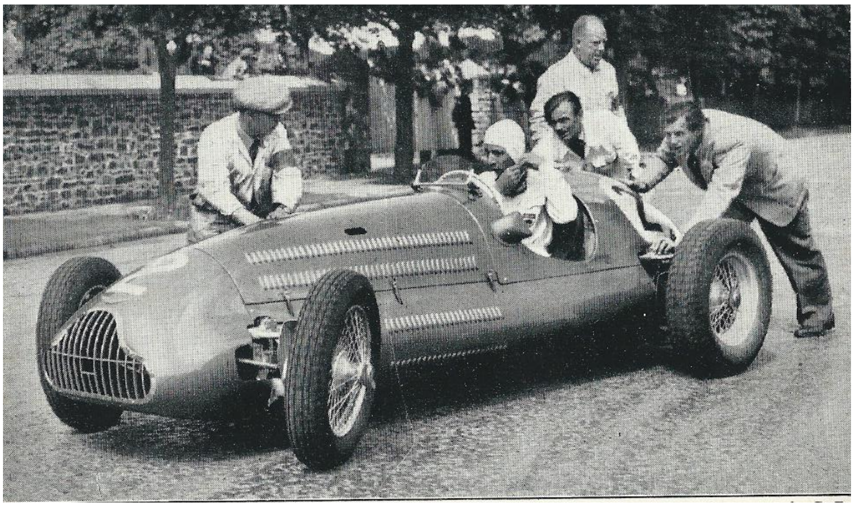
They were the Grand Prix Alta and the Formula F2 HWM.

Both, unlike the Cooper, were front-engined. Neither used roll-stiffening bars to control body roll.

Alta

The 1½ litre 4-cylinder supercharged Grand Prix Alta was designed by Geoffrey Taylor, owner of the small Tolworth firm, in late 1945. It was therefore the first car to follow the pioneering 1935 MG R-type suspension layout. The springing was novel, being rubber blocks in compression (supplied by Dunlop) – a method never seen again in racing cars, though adopted in 1959 by Alec Issigonis' BMC *Mini*.

The car first raced in 1948 and by 1950 three had been completed for low-budget private owners. Not surprisingly, the best result achieved by these was 2nd place in a minor Irish race in 1950.



DASO 779

The first appearance of the Grand Prix Alta in 1948.

Geoffrey Taylor is standing behind the car. The driver is George Abecassis and the man pushing on the right is John Heath. These two were partners in HW Motors and built the HWM-Alta in 1950, as described below.

The mechanic pushing the car is Alf Francis, later famous with HWM, Stirling Moss and Rob Walker.

HWM – Alta

John Heath, joint owner with George Abecassis of HW Motors of Walton-on-the-Thames, designed the 1950 HWM-Alta F2/Sports-racing car in 1950. The engine was a 2 litre 4-cylinder naturally-aspirated Alta designed by Geoffrey Taylor.

One transverse link of the pair required at each wheel was provided by a leaf spring.

The team of three cars were only raced stripped down for F2 so were handicapped by the 2-seater body. With Stirling Moss as effective No. 1 driver they achieved results 'way beyond what might have been expected from a low-budget operation. The most notable result was 3rd place by Moss in the 1950 Bari F1 GP behind two 159 Alfa Romeos.

Continued on P.2.

Stirling Moss driving the HWM in the 1950 “Daily Express” Trophy F1 race at Silverstone. He finished 6th in this F2 car.

Credit: unknown

Note 67

Siverstone Lap Speeds for 2.5Litre Grand Prix cars, 1954 – 1960

Best Lap Speeds by CoY on the 2.927 mile (4.711 km) circuit in practice

Year	Car	Driver	Time Min. Sec.	Speed MPH (kph)	Relative Speed
1954	Mercedes-Benz W196	Fangio	1.45*	100.35 (161.5)	Datum
1956	Lancia-Ferrari D50	"	1.42	103.30 (166.3)	+2.9%
1958	Ferrari 246	Hawthorn	1.40.4	104.95 (168.9)	+4.6%
"	Vanwall V254	Moss	1.39.4**	106.01 (170.6)	+5.6%
1960	Cooper-Climax T53	Brabham	1.34.6**	111.39 (179.3)	+11.0%

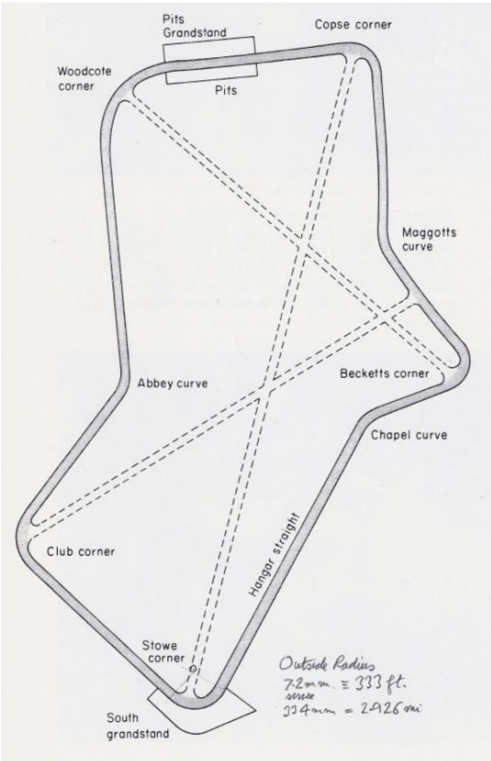
[In intervening years the British Grand Prix was run at Aintree]

*Corners only defined by drums at intervals. This hampered the W196 which had a fully-enveloping body. Fangio hit a drum during practice through inability to place the car accurately. Subsequently the drums were replaced by dwarf walls with the same trace which will have aided later lap speeds.
**Using Dunlop R5 tyres with nylon casings, worth 2% to 3% increase in lap speed versus previous R3 cotton-casing tyres (819).

On the same tyres, therefore, the mid-engined Cooper-Climax T53 with 240 HP was 5% faster than the front-engined Vanwall V254 with 265 HP 2 years earlier.

Silverstone circuit, 1954 – 1960

DASO 776



Note 68

The last front-engined Grand Prix car

The last front-engined Grand Prix car was the Ferguson P99 fitted with a 1.5L Coventry Climax FPF engine. It was also Four-Wheel-Drive (4WD).

The 1st race for this car was actually in one of the three 3 Litre Inter-Continental Series held in 1961 in the UK only, at Silverstone, fitted with a 2.5L Climax FPF. It failed to finish.

With the 1.5L Climax FPF Mk II engine installed it then competed in the 1961 British GP at Aintree driven by Fairman but, after having been push-started contrary to the rules when taken over by Moss, it was disqualified.

Later in 1961 Moss won the non-Championship F1 Oulton Gold cup race in wet conditions. This remains the only F1 race won by 4WD because several 4WD cars built in 1969 were unsuccessful and after 1982 the system was banned by FISA.

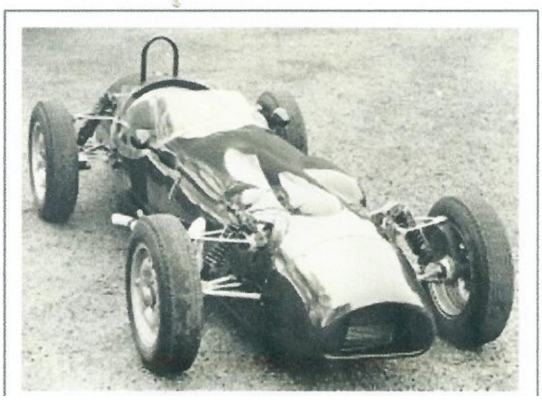
An interesting detail is that Ferguson tested the engine before and after the successful Oulton race, which was over 166 miles, at 152 BHP and 147 BHP respectively, both at 7,500 RPM. The loss was therefore only 3.3% (1049). Had the race been dry the car might not have been so competitive and the engine would no doubt have been given a harder time.

In 1963 in Antipodean races with the 2.5L Climax re-installed, it achieved a 2nd (Graham Hill) and two 3rds (Innes Ireland). A mountain climb in Switzerland (Ollon-Villars) was also entered driven by Jo Bonnier (result unknown).

In 1964 in the hands of Peter Westbury the 2.5L/P99 won the British Hill-Climb Championship.

Ferguson P99

DASO: The Ferguson Museum



Note 69

Oil scavenging

Vee-configuration engines with DOHC per bank have presented oil-scavenging problems at least since 1938 when the Mercedes M154 V12 had this trouble. In the 1939 M163 V12, with three pumps *supplying* oil to the crank and valve gear, six *scavenging* pumps were found necessary to return oil not only from front and rear of the sump but also from valve gear, superchargers, rear crank seal and the crankcase breather (30). The 120V6 Ferrari of 1961, as described by its designer Carlo Chiti (22), also suffered from oil-scavenging problems (which were described as 'barbotage' = splashing; the oldest engine lubrication system by splash was called in French 'graissage par barbotage', hence re-adoption of the term on the Continent for the new phenomenon). These problems were also met by increasing the number of scavenge pumps. Chiti (in the cited reference, published in 1980) credited Duckworth's DFV arrangements with the general solution to 'barbotage'.

An interesting numerical example of the power and lap speed losses due to oil churning with inadequate scavenging was the mid-1971 third redesign of the Matra 60V12 3L, B/S = 79.7/50 = 1.59. When first raced in the German GP at the Nurburgring, Amon's Matra with the new engine was 4.0% slower than pole (Stewart/Tyrrell-DFV) and 3.9% below the grid 2nd (Ickx/Ferrari 312B). Matra skipped the next race to investigate the poor power and found the cause as mentioned above (124), the peak being only 395 HP (981). When rectified the power rose to 460 HP (981). Amon then took pole for the Italian GP at Monza, 0.5% faster than Ickx, who was 2nd, both benefiting from slip streaming, and 1.2% faster than Stewart (who did not get a tow). Thus a reported 16.5% power rise translated into a 4.4% lap speed gain over the Ferrari (about a 4th root effect).

'Oil hiding', which is the same thing, ie oil not returning 'properly' above certain RPM, is known to have affected certain aero gas turbines in the '70s. It was thought to be associated with exceeding a critical Reynold's Number. It was noted that this effect, with great aeration, led to very high temperatures since the oil 'made many laps' before eventually reaching the reservoir. In these cases shrouding of certain gears effected a cure (880).

Note 70

Ferrari power

British constructors often queried Continental power quotes, not merely for the 1.4% smaller horsepower definition, and often with good reason (see Note 5 (Delage) and Note 6 (Maserati)). At the time that Ferrari were making the Dino V6 series these suspicions were very common. For instance 180 HP and even 190 HP were claimed for the original 65V6 Type 156 in 1957 but an experienced observer, Denis Jenkinson, thought 150 HP was nearer the mark (502). Later, In reviewing the 1961 – 1965 period he wrote that the 1961 redesigned 120V6 "*did in fact give 178 BHP @ 9,500 RPM on its first outing*" (824) – without quoting his "*factual*" source.

Carlo Chiti provided the "*Autocar*" magazine in late 1961 with a power curve showing (as translated) 192 BHP @9.500 Rpm (422). The two figures may be compatible with a season's development.

Walter Hassan believed that the 1961 Ferrari was 20 to 30 HP more powerful than his FPF Mk 2, which had just over 150 HP,i.e. 170 to 180 HP for the Italian car (515).

Note 71

Low pressure crankshafts

'Standard' crankshaft drillings

Where the lubrication of a crankshaft's big-end bearings begins with oil fed via the main journal bearings through radial drillings into longitudinally-drilled centre-line passages - the 'standard' arrangement* - the centrifugal pressure to be overcome at the main bearing surface is given by:-

$$OP = \frac{1.15.\ N^2.\ MJ^2}{10^7}\ \text{psi} \qquad \text{(ref 52, p.197)}$$

where N = RPM and MJ = Main Journal Bearing diameter (in inches), for oil of Specific Gravity 0.9.

For the 1977 Ford Cosworth DFV at 10,500 RPM and MJ = 2 3/8", OP = 72 psi. With the stated oil pressure of the 'standard' crank being 85 psi, this means the external oilways pressure drop was 13 psi.

'Low pressure' crankshaft drillings

Where the internal longitudinal passages are drilled off-centre at a radius RO, so as to shorten the radial drillings and reduce the 'centrifugal fling', the oil pressure required is:-

$$ROP = \frac{1.15.\ N^2.\ MJ^2}{10^7} \cdot \left(1 - \frac{(RO)^2}{(MJ/2)^2}\right)\ \text{psi}$$

This design was introduced into the 1977 Mg-alloy crankcase version of the DFV as noted in the main text and is stated to have reduced the necessary oil pressure to 60 psi. Assuming the external drop was 13 psi as before, then the crank requirement was 47 psi. This corresponds to:-

$$\frac{(RO)^2}{(MJ/2)^2} = 0.6 \qquad RO = 0.71"$$

With the overlap of main journals and crankpins being 0.88" = ((2.375" + 1.9375" - 2.550")/2), from an inner radius of 0.3" = ((2.550" - 1.9375")/2), the longitudinal drilling at 0.71" radius would penetrate directly into the crank pin to supply the inner end of the oil exit hole.

These details may not be correct for the particular DFV design of 'low pressure crankshaft' of 1977, but they indicate the powerful effect of the off-centre drilling.

Off-centre crank drilling was used first in the 1961 BRM V8 1.5L, according to (894), which includes a drawing of that crankshaft.

*There is another method, via double-end-feed into the crankshaft, eg in the Rolls-Royce Merlin 100 series aero engine of 1944, but this is not practical for most auto installations with a clutch on one end of the crankshaft.

Note 71B

Cylinder liners in compression

It was not the mixture of materials (cast iron liner and Al-alloy block) which caused the original Climax V8 problem but the length of liner *in compression* between the top and bottom locations. Given a warm-up time to stabilise the power readings the material differential expansion would have reduced some of the clamping load on the sealing ring *once per running cycle* from start-up to full speed but, presumably, there was still sufficient to seal the cylinder during ordinary test bed runs at incremental RPMs, when no trouble was experienced. This cycle would have occurred only a few times in an engines' racing life, after which the ring would be replaced at overhaul.

It was the in-cylinder temperature difference from throttle-open-to-shut cycling at the rapid rate of road racing, acting on the full length of the liner while the block at coolant temperature remained at expanded length, which caused the clamping load fluctuation leading to ring failure.

<u>BRM V16 problem</u>

A similar problem of cylinder sealing occurred in the 1947-designed BRM V16 which also had liners held in compression over their full length within the block by the detachable head (see Fig. N71B(B). In this engine, when the repeated liner expansions-and-contractions during the throttle cycling on the circuit forced the Al-alloy block locations to give way in fatigue, the gap created at the top spigot joint in the head allowed the coolant to enter the tiny cylinder on shut-throttle and form an hydraulic lock (838). The resultant cast-iron liner breakage led to such disastrous secondary damage that it took a long time to determine the primary failure (56).

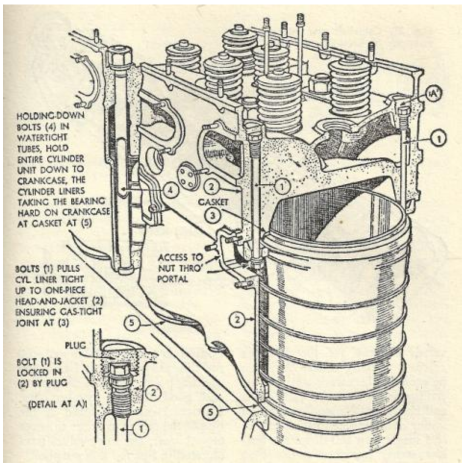
Not surprisingly, when BRM designed their IL4 2.5L NA engine for the 1954 Formula they went to liners screwed into the head, as had been done by Colombo in the 1938 MSC Alfa Romeo 158 and in the Lampredi-designed 1950 – 1953 V12 and IL4 NA Ferraris.

<u>Rolls-Royce *Merlin* experience</u>

The racing engine designers were in good company in falling into the trap of cylinder liners retained in compression. Rolls-Royce built the same thing into the *Merlin* in 1937 when the Mk II was hastily fitted with the ex-*Kestre*l layout (which leaked (900)) after the Mk I "ramp head" engine proved unreliable (see Fig. N71B(A). It was not until 1942 when the "two-piece-block" having a top-flange-located liner finally went into production (initially at Packard in the USA) that the cylinder leakage problem at high boost in fighter duty was overcome.

Ironically, the Mk I had a top-flange-liner-location!

Fig.N71B (A)
The Rolls-Royce *Merlin* Mk II
cylinder layout.
DASO 900



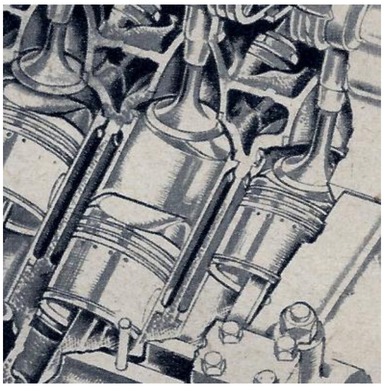


Fig. N71B (B)
The BRM Type 15
cylinder layout.
Motor 25 Nov 1953

It is curious that Erich Richter, who did the detailed BRM design and had worked at Rolls-Royce previously, did not use the liner solution adopted there,

Note 72

Power Disinformation

Note 2 refers to ***disinformation*** on power to dishearten rivals. Before practice for the 1966 Italian GP Jack Brabham pulled the legs of his competitors by unloading from the truck a new REPCO 620 V8 engine in a crate stencilled “Monza 350 HP”. In fact it had given about the usual 300 HP and attempts to secure more with a higher compression ratio had resulted in a piston failure (842)!

However, his *300* horses were certainly present and willing – in the race Brabham retired when leading due to oil leaking from a lost timing-chain inspection plate (a preparation fault). No replenishment was permitted by the rules, even if the plate could have been replaced.

The other Brabham (Denny Hulme) was 3rd behind 2 new 36-valve Ferrari V12s.

Note 73

Cosworth SCA v. Repco 740

These two engines had similar cylinder head/piston designs, with the combustion chamber formed in the piston top mostly by the valve clearance pockets (the SCA also had a pocket just under the sparking plug). Heads were flat with two 'vertica1' valves operated by SOHC. The same-side inlet and exhaust porting was also similar with the inlets at 40° or 45° to the cylinder axis (ie 50^0 to 45° downdraught). The SCA was designed for F2 in 1963, was raced on carburettors in 1964 and port fuel injection in 1965. The comparison with the 1967 Repco 740, also fuel-injected on petrol, is as follows:-

		1965 Cosworth SCA	1967 Repco 740	
Data sources		63,583	37,842,844	
Configuration		IL4	90V8	
B/S	mm	80.97 / 48.41 = 1.673	88.9 / 60.325 = 1.474	
V	cc	997	2996	
R		12.5	11	
IVA/PA		0.207	0.216	Note A
CRL/S		2.85	2.65	Note B
PP	HP	140	300	
@NP		10,250	8,000	
PP/V	HP/L	140.4	110.2	Notes C and D
BMPP	Bar	12.26	12.32	Note E
@MPSP	m/s	16.54	16.09	
MGVP	m/s	79.90	74.49	

Notes

A. It is *presumed* that valve sizes in the 740 were the same as the Repco 620. A 'big-valve' engine was tried but was unsuccessful (846) for unknown reasons. The space occupied by the valves, relative to that available, was:-

(IVD + EVD/B	84.7%	85.7%

B. It is *presumed* that the 740 CRL was the same as the 620.

C. As BMPP and MPSP were very similar it follows that the 27% advantage in PP/V to the SCA was basically because 1/S was 25% greater, as explained in the main text.

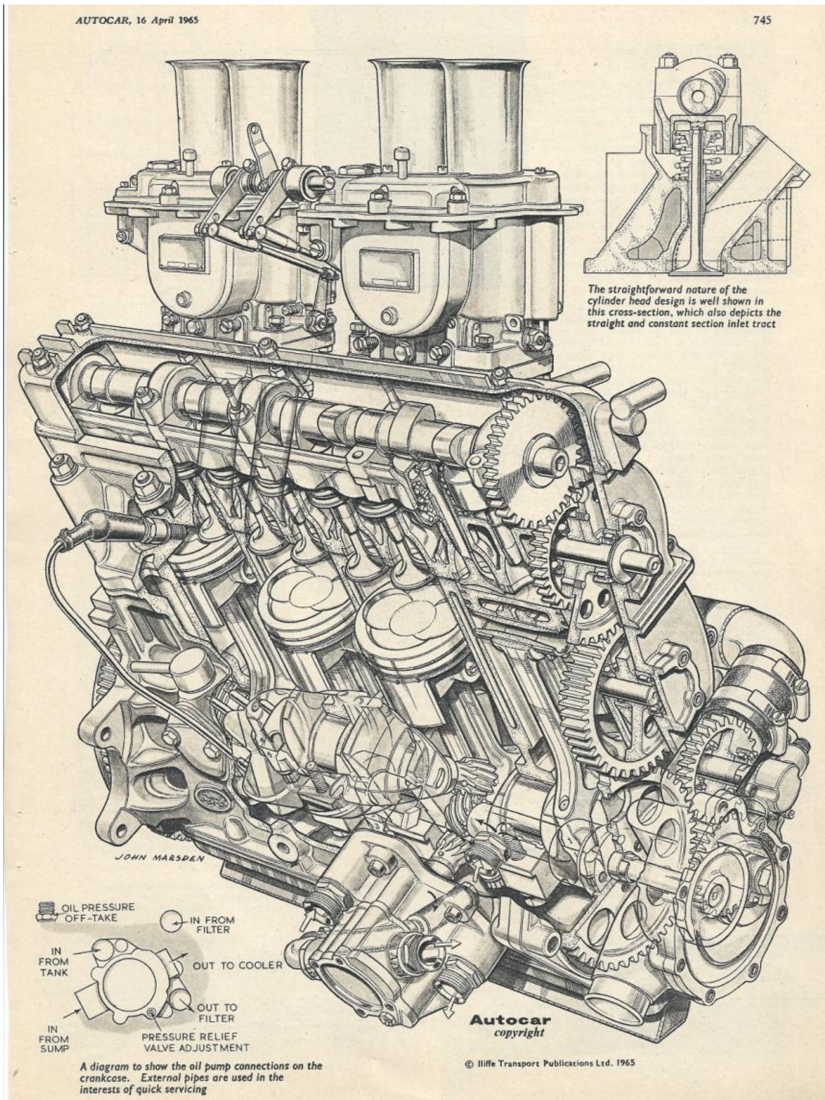
D. Ref (884), p.67, quotes from Judd's correspondence with Repco that he had learnt in October 1966 that Cosworth applied a humidity correction to test bed power as well as the usual pressure and temperature corrections. It was estimated that the 140 HP quoted for the SCA would be about 123 without that humidity adjustment. This author feels that, if Cosworth applied a humidity correction (which undoubtedly is a factor in output), then it was justified.

The concern which Repco were then showing over their PP/V v the SCA seems to have overlooked the 1/S factor shown above.

E. Duckworth was dissatisfied with the slow burning in the SCA, which required an ignition advance of 49° BTDC (60). Perhaps this was due to the 'chamber-in-piston'

design, which also meant increased mass and greater heat rejection that had to pass via the rings and under-crown oil splash. However, Tony Rudd had found in his BRM F2 IL4 1.0L engine, apparently after 1966, that shortening the con rods by 0.5" to reduce CRL/S from 1.91 to 1.7 (-11%) had raised power from 128 HP to 136 (+6%) (40). He deduced that faster burning had more than offset increased piston friction. It is possible that the extreme CRL/S of the SCA had affected its performance.

The 1964 type SCA engine with 2 x Weber 40DCM2 carburetters is illustrated below.
This was rated at 122 HP @ 9,000 RPM (63).



DASO 63

Note the "Bowl-in-crown" combustion chamber.
The cam shape reflects the then-standard Cosworth racing-engine timing of:-
Inlet opens 58^0 BTDC; closes 82^0 ABDC:
Exhaust opens 82^0 BBDC; closes 58^0 ATDC.
Inlet Open Duration (IOD) = Exhaust Open Duration (EOD) = 320^0:
Overlap (OL) = 116^0.

The engine was installed with the inlet tracts vertical.

Note 74

Top speeds, 1960 to 1966

Ref (845) gives Brabham's experience of the top speeds at Spa in 1966 compared to 1960 which, he said, "*surprised and disappointed us*":-

Date	Car	Engine	Power HP	Top speed MPH	Lap speed MPH
1960	Cooper T53	Climax FPF 2.5L	240	178	137.1
1966	Brabham BT19	REPCO 620 3L	300	172	142.2
1966 v. 1960			+25%	-3.4%	+3.7%

This effect was attributed to the much-wider high-hysteresis tyres which had been developed by Dunlop to raise cornering speeds and hence lap speeds, despite the drop in top speed. Rear tyre tread widths had doubled – from 5" to 10" – and the tread compounds were now synthetic instead of natural rubber.

The same effect had been remarked upon by Colin Chapman in describing the development of the 1.5L Lotus (see under the 1.5L Formula Summary, 2nd Naturally-Aspirated Era (2NA) Part 3 at PP 20-21).

However, it may be that the 1966 Brabham drag was also worse than the Cooper in 1960 for the reason speculated regarding the V8 exhaust system in Eg. 46, compared with the simpler system of the IL4 FPF.

NOTE 75 Rivals to the Ford-Cosworth DFV

The 10 major engine makes against which the Ford-Cosworth DFV competed from Mid 1967 to the end of 1983 and their main specifications are listed below.
DWC = Drivers' World Championship; CWC = Constructors' World Championship.
TC = TurboCharged; all others were 3L Normally Aspirated.

Configuration	B/S mm	Valve No. per Cylinder	VIA^0	Dates : from	to
FERRARI	**7 Specifications**	**47 Wins (1968 on)**	**DWC 1975, 1977, 1979**	**CWC 1975, 1976,1977,1979 1982, 1983**	
•60V12 Outside Exh.	77 / 53.5 = 1.439	3	Wide (Axial Inlets)	Monza 1966	April 1967
•60V12 Central Exh.	As Above	3	Wide (Axial Inlets)	April 1967	Monza 1967
•60V12 Central Exh.	As Above	4	Narrow	Monza 1967	Through 1968
•60V12 Outside Exh.	As Above	4	Narrow	1969	Through 1969
•F12	78.5 / 51.5 = 1.524	4	20	1970	1971
•F12	80 / 49.6 = 1.613	4	Narrow, prob. 20	1972	Early 1981
•120V6 1.5L TC	81 / 48.4 = 1.674	4	38	Early 1981	Past 1983
REPCO	**2 Specifications**	**3 Wins (In 1967 after DFV debut)**	**DWC 1967**	**CWC 1967**	WCs incl. 1 Win for 1966 Spec. prior to DFV debut
•90V8 Central Exh.	88.9 / (1982 on) 60.325 =1.474	2	0	Through 1967	
•90V8	As Above	4	30	Through 1968	
Eagle-Weslake		**1 Win (1967 after DFV debut)**			
•60V12	72.8 / 60 = 1.213	4	30	Monza 1966	Part 1968
Honda	**3 Specifications**	**1 Win (1967 after DFV debut)**			
•90V12 Central Exh.	78 / 52.1 = 1.497	4	Wide (Axial Inlets)	Monza 1966	1967
•?V12 Outside Exh.	?	4	Not Wide	Through 1968	
•120V8 Air-Cooled	88 / 61.4 = 1.433	4	Not known	1968 (1 race)	
Maserati	**2 Specifications**	**0 Wins after DFV debut**			
•60V12	70.4 / 64 = 1.10	2	78 (Axial Inlets)		1967 Pre-Monza
•60V12	75.2 / 56 = 1.343	3	0	Monaco 1967	End 1967
BRM	**6 Specifications**	**4 Wins (1971-1972)**			
•I 16	69.85 / 48.895 = 1.429	2	52 (Axial Inlets)	Mid 1966	Start 1968
•I 16 Mk2	As Above	4	13	Not raced	
•60V12	74.6125 / 57.1 5 = 1.306	2	60 (Axial Inlets)	Through 1968	
•60V12 Central Exh.	As Above	4	13	Through 1969	
•60V12 Outside Exh.	As Above	4	13	Through 1970	
•60V12	78.5 / 51.5 = 1.524	4	13	1971	1977
Matra	**3 Specifications**	**3 Wins (1977,1978)**			
•60V12	79.7 / 50 = 1.594	4	56 (Axial Inlets)	Through 1968	
•60V12	As Above	4	33	Through 1970	
•60V12	As Above	4	15	1971	1972
				1976	1978
				1981	1982
Alfa Romeo	**4 Specifications**	**2 Wins (1978)**			
•90V8	86 / 64.4 = 1.335	4	39.5	1970	1971
•F12	77 / 53.6 = 1.437	4	35	1976	1978
•60V12	78.5 / 51.5 = 1.524	4	35	Start 1979	Mid 1979
•90V8 1.5L TC	74 / 43.5 = 1.701	4	Not Known	Late 1982	Past 1983
Renault		**15 Wins (1979 on)**			
•90V6 1.5L TC	86 / 42.8 = 2.009	4	21.5	Mid 1977	Past 1983
BMW		**5 Wins (1982 on)**	**DWC 1983**		
•IL4 1.5L TC	89.2 / 60 = 1.487	4	40	Mid 1981	Past 1983

NOTE 75 Continued

While the above Table shows the serious competition which the Ford-Cosworth DFV met and defeated in 2 races out of every 3 over 1967–1983, there was one famous name in racing which did *not* enter the lists against it. It will never be known if Mercedes-Benz *could* have beaten the DFV consistently if they *had* built a new racing car in, say, 1978. As they described the many small successful chassis-builders with Ford-Cosworth engines as “boutiques”, it may be that they did not care to race against them for fear of any victories being ascribed simply to their vast resources and any defeats making them look foolish. What we *do* know is that in 1980, when they wanted a high-output rally engine for their 4-cylinder type 190 production saloon and an in-house redesign was not sufficiently powerful, they contracted with Cosworth to make for them a 16-valve head (Cosworth type WAA) which did meet their requirement and later engaged them to supply a modified road-going version in quantity (468).

Note 75B

DFV swept volume

Cosworth's own data sheets (191) gave B = 3.373" (85.6742 mm) and S = 2.550" (64.77) but then put swept volume at 182.64 cubic inches (2993 cc) when the listed B & S correspond to 182.285 cubic inches (2987.1 cc). No explanation is known for this. The corresponding data sheets for the DFX TC variant, with the same B and S = 2.256" (57.3024) specified the swept volume correctly as 161.27 cubic inches (2642.7 cc).

Note 76

Pressure-charging by exhaust turbine

When a piston engine is Pressure-Charged (PC) mechanically, power is taken directly from the crank to drive the inlet supercharger. However, because the inlet pressure in the cylinder is greater than that available with Natural Aspiration (NA) (after allowing for the pressure loss through the valve and the valve opening before TDC) the crank no longer has to supply power to the piston during the induction stroke but is provided with power. As there are mechanical and aerodynamic inefficiencies in the system this recovered power is substantially less than that subtracted originally, but it is some reduction of the power cost necessary to raise the Manifold Density Ratio (MDR) above ambient, which itself multiplies power. The Mechanical Supercharger (MSC), therefore, lowers the Mechanical Efficiency (EM) of the engine compared to an NA unit.

When the power to drive the supercharger is taken by a turbine from the high-temperature exhaust flow, the back pressure in the cylinder rises above the NA level during the exhaust stroke and reduces the crank output but the inlet pressure once again supplies power to the crank and, with typical turbine and compressor efficiencies, this is equal to or greater than the subtraction because of the difference in temperatures. Overall the crank output by TurboCharging (TC) is greater for this reason, compared to NA, before adding the gain from a higher MDR. The value of EM is raised in this case (but this diminishes as IVP is increased).

In effect a gas turbine consisting of a compressor, a combustion chamber (the piston engine) and a turbine has been coupled pneumatically to the piston engine. As the rules allow only one engine per car this is a breach of those rules.

Without elaborating on the above details, Keith Duckworth put forward the argument regarding illegality to the FISA, after TC engines were allowed into competition with his NA 3L DFV, simply on the basis that the TC turbine was another engine and an appropriate equivalent amount should be subtracted from the 1500 cc of the piston engine (851), but they ignored this argument*.

It is interesting that from 1977 Cosworth were making the TC DFX engine to the limited-swept-volume rules (2.65L) of Indy car races without worrying about any theoretical objections but, of course, there was no NA alternative.

Eventually, to stay in GP racing, Cosworth designed, and Ford USA financed, a 120V6 1.5L TC engine (type GB) for 1986, after they had been assured by Balestre of FISA that the regulations would not be changed before the end of 1990. The GB was up to competitive power but was not fully developed before the regulations were changed to ban TC after 1988, ostensibly to reduce cost (TC powers had already been restricted by fuel limits to reduce speeds). Therefore, the Cosworth GB programme was discontinued at the end of 1987.

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*The attitude of FISA to the DFV and TurboCharging *may* have been that of a legendary umpire in allowing a blatantly-false LBW appeal against a batsman who had been at the crease very successfully for a long time and who then expostulated "*That wasn't out!*". The reply was "*Tha's batted long enough!*".

Note 77

Cosworth's investment in a dynamometer

The size of financial risk, which Duckworth took in 1959 when company money was very scarce by borrowing £600 to buy a dynamometer so as to develop engines scientifically, can be judged by noting that each 1960 conversion of a Ford 105E engine for Formula Junior racing was sold for £145 (60), of which perhaps 10% at the most was profit after tax and before paying back the loan with interest. That is, approaching 50 conversions had to be sold before there was any cash surplus. However, the engine was so successful that 126 were supplied to Lotus in 1960 (60). The company's future was, therefore, ensured.

Note 78

The four valves-per-cylinder revival, 1959 onwards

Post WW2, no GP CoY had more than two valves-per-cylinder (2 v/c) until 1965 (the Coventry Climax FWMV Mk 6) but a renewed interest in four valves-per-cylinder (4 v/c) for GP engines had begun in 1961. It was not a coincidence that this was the year that the 4 v/c Honda RC144 IL2 125cc and RC162 IL4 250cc motor-cycles won Championships in those capacity classes. These 1961 Hondas were the first 4 v/c engines to win classic motor-cycle races since Rudge-Whitworth in 1934. Honda had entered international racing in 1959 with a 4 v/c RC142 IL2 air-cooled 125cc, which was reliable but not fast enough. They produced an improved 125cc (RC143) the following year plus, effectively a doubled-up 250cc version (RC161), both of which showed promise which was confirmed in 1961 after more development. The 1961 250cc, designed by Yoshio Nakamura, had B/S = 44/41 = 1.07 and VIA = 76°, a figure required to get adequate air-cooling to the top of the cylinder head, in which squish plateaus were incorporated. BMPP was 11.5 Bar at MPSP = 19.1 m/s with MVSP = about 2.9 m/s. R x VIA = 10.5 x 76° = 798° and MGVP = 48.7 m/s. None of these factors was exceptional. Honda's success was due to the smaller stroke of the 125 twin and 250 four in competition with 125 singles and 250 twins, plus careful and scientific attention to combustion chamber shape and exhaust scavenging (with low-taper megaphones) to offset the adverse higher surface area/volume ratio of the smaller cylinders, along with efforts to minimise the loss of mechanical efficiency at high RPM (75). Honda continued from 1961 through 1967 to produce ever-higher HP/litre in all classes in the classic way of reducing stroke, mainly by miniaturising cylinders, except for their 500cc engine, rather than raising B/S. Their success in this is illustrated on an attached Figure 115/dst. Every engine up to 1967 was 4 v/c. (To complete a description of the Honda four-stroke GP motor-cycle series, the 1984 NR500 90V4 500cc engine has been added, which had 8 v/c in 'race-track' shaped ('oval') cylinders, the equivalent-circular B/S being 75.36/28 = 2.69. This was an attempt, beginning in 1979, to wrest back racing supremacy from the two-strokes which had dominated all motor-cycle GP classes since 1975 but it failed despite achieving 268 HP/L. Its influence on GP cars was that non-circular cylinders were banned!).

Chronologically in the post-WW2 automobile 4 v/c story there were several abortive designs and one success:-

- In early 1952 Mercedes-Benz tested a 4 v/c single-cylinder unit, experienced valve-gear problems (curiously) and then adopted a 2 *vlc* desmodromic head instead for the M196 (468) - described in Eg 32;
- Stewart Tresilian designed an IL4 2L of high *B/S* ratio with 4 v/c for Connaught, who did not have the money to build it (587);
- In 1954, Tresilian then produced a similar IL4 2.5L design for BRM, the extreme B/S = 102.87 / 74.93 = 1.37 being matched to the 4 v/c head (587). However, Peter Berthon altered this to 2 v/c with hollow-headed valves - and incurred much valve and valve-spring trouble;
- In consequence of their 2.5L 2 v/c problems BRM did try 4 v/c in a single-cylinder test unit 'some years earlier' than 1964, ie pre-1961 (when the formula swept-volume changed) but the result did not encourage them to proceed to a main engine (587, 830);
- There was a *successful* sports-racing Borgward RS IL4 1.5L engine with 4 v/c at VIA = 64° in 1958 which, in a Cooper chassis, secured seven F2 wins in 1959. Details of this unit are given in Note 79.

Ferrari was the first GP engine maker during the 1.5L formula to consider 4 v/c as a way of extracting higher power and, pre-season, announced that his 1962 car would have this feature

in a redesigned 120V6. The engine was built and shown to the Press in a car but it never appeared in competition. Coventry Climax designed a 4 v/c head for their 90V8 in 1963, as described in Eg 44, taking a long time to obtain more power than their 2 v/c but eventually racing it in 1965. BRM designed another 4 v/c head for their 90V8 at the same date; it never exceeded the 2 v/c power and was discarded (830). The Climax and BRM designs had relatively wide VIA: 60° for the former, 68° for the latter (836), with some squish for the Climax but none for the BRM, which had axial ('downdraught') inlet ports.

Honda entered GP car racing in 1964 with the RA271 60V12 1.5L (B/S = 58.1 / 47 = 1.24), having their 'trademark' 4 v/c wide-angle head (VIA = 65°), the engine mounted transversely. This was improved to the RA272 in 1965 and this won the final race of the 1.5L formula.

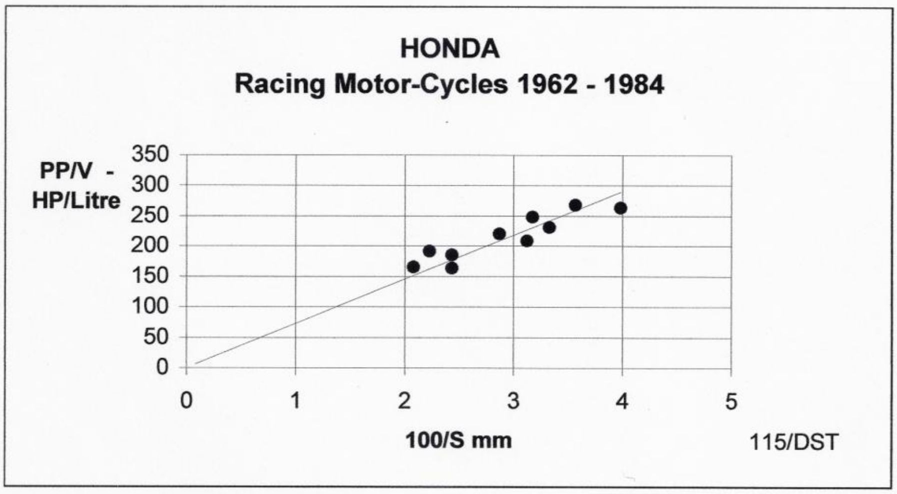
Honda also provided in 1965 a F2 IL4 1.0L engine for a Brabham chassis, once again with 4 v/c wide-angle head, but this needed a short-stroke redesign for 1966, after which it was very successful, the Brabham-Honda winning the F2 Championship easily in that year. Part of the reason for this success, following two F2 Championships won by the SOHC Cosworth SCA, was that Keith Duckworth's efforts that year were devoted to a design for the new 1.6L F2 formula to come into effect in 1967. He spent 1965 first choosing a suitable cylinder head for the Ford Cortina 120E production block and then designing in detail his own narrow-angle (VIA = 40°) version of a 4 v/c engine, the FVA ('Four Valve type A'), which is described fully in the main text, Eg 47 et seq, and Note 79.

While the Cosworth FVA became a very successful commercial racing engine, powering all the 1.6L F2 Champions bar one in 1967-1971 and forming the top-end basis of the phenomenally successful DFV GP engine (qv Eg 47), which first raced and won in June 1967, it has to be recorded that it was preceded in 1964 by the Shell-financed Weslake WRP22 IL2 375cc research unit with narrow-angle (VIA = 32°) 4 v/c head (587). This produced BMPP = 11.6 Bar at MPSP = 20.7 m/s on petrol. Enlarged to 500cc with VIA = 30° it was developed to BMPP = 14.3 Bar at MPSP = 20.0 m/s (587), and this was the basis of the Gurney-Weslake or 'Eagle' 60V12 3L GP engine which first raced in September 1966. The Weslake WRP22, therefore, was the first of the second phase narrow-VIA engines which really produced the 'Four Valve Renaissance' (in the late Brian Lovell's words) where the first phase had begun in 1959 with the wide-VIA Honda (but see also Note 80).

HONDA
Racing Motor-Cycles 1962 - 1984

Example						SO18	
DASO	14,228, 354	14,228, 354,357	14,228, 354	14,228, 354	14,228, 354,357	14,228, 354	14,228, 354,357
Year	1962	1962	1963	1965	1965	1965.5	1966
Model	RC145	RC163	RC146	RC148	RC165	RC149	RC180
100/S mm	2.439	2.439	3.125	3.333	2.874	3.984	2.083
PP/V HP/Litre	163.6	184.5	208.7	230.6	220.5	263.2	165.5

DASO	14,228, 354,357	14,228, 354,357	16,74,228, 354,357,358
Year	1967	1967	1984
Model	RC168	RC181	NR500
100/S mm	3.175	2.227	3.571
PP/V HP/Litre	248.5	191.3	268.2



Note 79
Comparison of Climax FPF with Cosworth FVA

			Other comparisons	
Engine	Coventry Climax FPF L5L Mk 2	Cosworth FVA	Coventry Climax FWMV Mk 6	Borgward RS
Data sources	33,54,56,57, 131	58, 63, 247, 583	34	181, 205, 711
Date	Mid 1961	Mid 1967 (Note A)	1965	1958
CN	IL4	IL4	90V8	IL4
B/S mm	81.788/71.12 = 1.15	85.725/69.14 = 1.24	72.39/45.466 = 1.592	80/74 - 1.081
V cc	1495	1596 +6.8%	1497	1488
Fuel	Petrol 102 RON	Petrol 102 RON	Petrol 102 RON	Petrol 102 RON
R	10.7	11	12	10.2
VNI, VIA	1,66°	2,40° (See Note 78)	2,60°	2, 64°
RxVIA	706°	440°	720°	653°
IVA/PA	0.296	0.305	0.266	0.340
Valve gear	DOHC, CVRS	DOHC, CVRS	DOHC, CVRS	DOHC, CVRS
IVL / IVD	0.234	0.304 +29.9%	0.318 -4.4%*	0.227 +33.9%**
IOD	290°	320° +10.3%	290° +10.3%	294° +8.8%
LIN mm	?	297 (Note B)	301 (Note B)	?
Inlet downdraught angle	12°	30°	20°	6°
Circumferential swirl	Yes	No	Yes	No
Tumble swirl angle (Note C)	0	20° (See Note 26)	0	0
Squish	No	Yes	Yes	No
Fuel supply	2x2 choke Weber carbs	Lucas Mk 2 fuel injection	Lucas fuel injection	Bosch direct fuel injection
Ignition	2 plugs/cyl, magneto	1 plug/cyl, Lucas transistorised	1 plug/cyl, Lucas transistorised	2 plugs/cyl/coils
PP @NP HP @ RPM	151 @ 7500	222 @ 9000 +47% (Note D)	212 @ 10300 +4.7%	150 @ 7500 +48% (Note I)
MaxRPM	8200	9500	11000	8500
PP/V HP/litre	101.0	139.0 +37.6%	141.6 -1.8%*	100.8 +37.9%**
BMPP Bar	12.05	13.83 +14.8%	12.30 +12.4%	12.03 +15.0%
MPSP m/s	17.78	20.74 +16.6%	15.61 . +32.9%	18.50 +12.1%
MVSP m/s	3.23	3.44 +6.5%	3.58 »3.9%	2.. 30 +49.6%
MGVP m/s	60.1	68.0 +13.1%	58.7 +15.8%	54.41 +25.0%
MPDP g	2862	4009 +40.0% (Note G)	3270 +22.6%	2941 +36.3%
W kg	129 (Note E)	118 (Note F) -8.4%	135 -12.6%	128 -7.8%
HP/W HP/kg	1.17	1.88 +60.7%	1.57 +19.7%	1.17 +60.7%
Price	£1500 (£1830 @ 1967 level)	£2500 (Note H)	£5000 (£5300 @ 1967 level)	?
Price/HP@ mid 1967 level	12.1	11.3 -6.6%	25.1 -55.0%	
			**** FVA relative to FWMV 6***	***** FVA relative to RS***

Notes

(A) Raced in F2 over 1967-1971. During development the engine was raced by Mike Costin in a Brabham chassis at club meetings from July 1966 (as the FVB short-stroke variant).

(B) Note 27 indicates that the inlets would resonate at the following MPS:

FVA	88.25 x (69.14/297) = 20.5 m/s
FWMV6	88.25 x (45.466/301) = 13.3 m/s

(C) Angle which the outer wall of the inlet port just before the valve seat makes with the valve centreline.

(D) In early 1967 Cosworth built the FVB engine, an FVA destroked to 1.5L, to check the forthcoming DFV output. This gave 200 HP; (134 HP/L) (605). In 1969 the FVA power was raised by:- modified valve timing; altered port shapes; 4-into-l exhaust system. Con rods were strengthened for higher RPM. The 1970 rating was therefore:- 240 HP @ 9,500 RPM; (150.4 HP/L) (168), equal to BMPP = 14.16 Bar @ MPSP = 21.89 m/s.

(E) All Al-alloy static structure: 115.7 kg for Mk 1 (33) plus 13.6 kg for Mk 2 (56) with 2.5L-type crankcase.

(F) A production cylinder block was required by F2 rules. The part chosen was Ford 120E, cast iron, bored-out from 3 3/16" to 3 3/8". Al-alloy head. The Lucas 100 psi f.i. pump and Lucas spark generator were both chassis-mounted and not included in engine weight. If these items had been counted the weight would have been similar to the FPF Mk 2.

(G) A Dykes top compression ring was used on both engines, but the FVA made full use of this whereas Walter Hassan was still timing his engines to avoid exceeding 100,000 ft/sec^2 (3100g). See Note 13 Part II.

(H) Development costs charged to other accounts in each case. Climax absorbed theirs, Cosworth's were paid by Ford.

The production iron block would have reduced the price of the FVA.

(I) 150 HP was quoted as sustained power, but up to 165 HP for sprints (BMPP = 13.23 Bar). This might be compared with the developed 240 HP of the FVA (Note D) at BMPP = 14.16 Bar, which was 7% greater.

The Borgward RS had no deliberate shaping of the inlet ports to create ‘Barrel Turbulence’ (Tumble Swirl).

Note 80

The FVA and DFV and 'Tumble Swirl'

It is not surprising that the initial published descriptions of the FVA and DFV did not mention 'Tumble Swirl' (TS). However, it is significant that, in various interviews with Keith Duckworth (KD) over the years of front-line service by the engines the words 'Tumble Swirl' never crossed his lips (eg January 1970 (853), early 1971 (60), July 1982 (850, 851)). In particular, ref (60) has a frank exposition of his design approach plus a review of DFV development to early 1971, but KD says of the cylinder head (port shapes) only that he wished to keep his philosophy to himself as he thought it obvious that most people had not thought the problem through. Ref (853) states that KD always made the prototypes of a new inlet port/combustion chamber shape himself, with hand tools. The actual parts of the engines were not then secret, of course, since Cosworth sold them to all comers - egs the Japanese trading company Mitsui bought a pair of DFVs in 1968, and Mercedes-Benz bought one in 1969 (60). Undoubtedly, every engine maker obtained one in some way over the years. They would have been failing in their duty if they did not!

There was a detailed description of the FVA in the March 1968 *Automobile Engineer* (583)- almost certainly at Ford's request - which gave a section of the engine with the crucial inlet port shape needed to optimise TS - for those with eyes to see (see Note 26) - but TS was not mentioned in the text. This author is not aware of any published section of the DFV - in fact its debut, after the early descriptions, marks the demise of the detailed analyses of racing engines which used to be common in the *Motor* and *Autocar* magazines.

This 'black-out' on TS leads this author to conclude that it was probably the most important feature in producing the superior FVA and DFV performances, and KD did not wish this to be known.

It is not clear when TS was recognised generally as a critical element in obtaining high BMEP (> 13 Bar) at high MPS (> 20 m/s). The 1964 (pre-FVA design) Weslake 4 v/c Shell 375 cc twin did not have the necessary port shape (836). Of post-FVA racing engines where cross- sections have been published, the 1970 Ferrari 312B did (187), the 1976 Renault CHI F2 did not have the 'TS shape' (485), nor did the destroked TurboCharged (TC) version, the 1977 EFl (638). The Honda RA168E, dominant in the last TC season of 1988, did not have it (20) but the 1992 Honda RA122E/B did (69). The 1991 Mercedes M291 sports racing engine did not - in fact, it reverted to the axial inlet port, which prevents TS (468) (see Note 26).

In August 1997 Ian Bamsey, Editor of *Race Tech* magazine, interviewed Nick Hayes, then F1 Programme Director of Cosworth, and put it to him that TS was *"an integral part of the concept"* of the DFV. Hayes accepted that (419).

Note 80B

"Barrel Turbulence" aka "Tumble Swirl"

Since writing Note 80 the author has been told by Michael Costin that Keith Duckworth used the term "Barrel Turbulence" to describe the in-cylinder effect which he pioneered and which was later called by others "Tumble Swirl".

P.S. 6 September 2019.

Correspondent Ron Rex has pointed out to the author recently that a written description which *could* have led a knowledgeable reader to understand Barrel Turbulence appeared in *Motor* 12 January 1974 (DASO 206) which described the new Cosworth type GAA engine. This included information supplied by the designer, Mike Hall. As published the crucial part read:- "*...the straight-line flow of gas through the upper parts of the valves creates a vertical swirling motion which ensures good mixing...*". Missing was the important next step, that on the compression stroke conservation of angular momentum then creates a much fiercer vortex before ignition occurs. Whether Keith Duckworth had authorised this release is unknown.

Note 81

Ford-Cosworth big-end journal diameters

It is an interesting design point that Duckworth was able to retain a 1 15/16" (49.2 mm) big-end journal diameter unchanged from the 1959 production Ford 'New Anglia' 105E engine* in all his engines from the 105E FJ conversion through the SCA and FVA into the 1967 DFV. It then stayed unchanged in the DFV as it was developed during F3000 racing (at a regulation 9000 RPM) until, in 1992, it was reduced to 1 25/32″ (45.2 mm) to cut bearing friction. This was in parallel with similar smaller crank diameter on Cosworth front-line GP engines and, together with DFV main bearings reduced from 2 3/8″ (60.3 mm) to 2 1/8″ (54 mm), pin and journal averaging 9% reduced diameters, gained 2% of power (65). Materials of the crank had, of course, been improved: the spherical-graphite hollow cast iron 105E crank had been satisfactory for the FJ engines, but - from the SCA onward - nitrided En40B (3.5% Cr, 0.5% Mo, 0.5% Mn, 0.25% C, etc, 95% Fe) had been used, ie from a 120 HP to a 500 HP crank, all around 10,000 RPM, on that 1 15/16″ pin.

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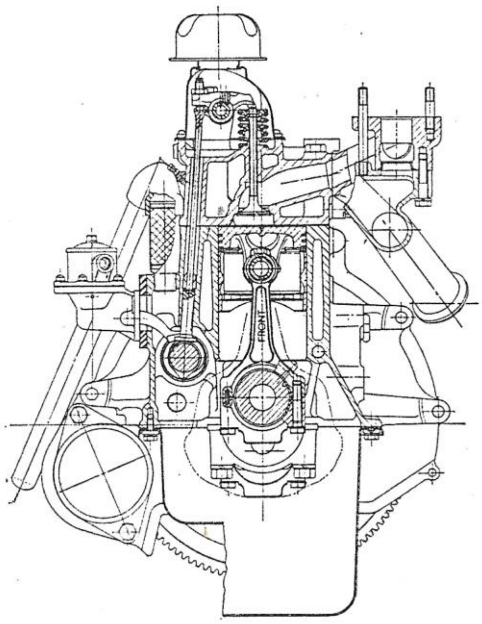
* The original 105E crank had been 1 3/4" in big-end diameter, but this had cracked during development testing and was changed to 1 15/16" (156).

Fig.N81A
1959 Ford 105E
IL4 3 3/16''/1.29/32'' = 1.672 60.85 cid
(80.96mm/48.42 997 cc)
39 HP @ 5,000 RPM
DASO 156



Note 82

Mercedes-Benz/Bosch experiments with RPM governor

Ref (468) describes how Bosch designed an RPM-limiter to cut the ignition of the 1938 Mercedes M154, which was tested on a rolling-road test bed and in a practice session. It was variable in its effect and so was put aside. Like many vehicle electrical items, it worked on the test bed, was erratic in the field and worked again when re-tested on the bed! Differences in mountings, temperatures and vibration are often found ultimately to be the cause or causes of such irritating and time-consuming behaviour.

When they returned to racing after WW2, the Mercedes M196 positively-controlled valve-gear made a governor unnecessary.

Note 83

Exhaust resonant speed

In the '60s Jack Williams of AJS gave the following relation for the crank speed at which the engine's exhaust system would resonate (to improve breathing via overlap). He had found that it fitted accurately the case of his type 7R one-cylinder 350 cc motor-cycle racing unit (544).

$$\text{Resonant } N = \frac{1}{24} \times \left(\frac{EOD}{(LEX + LX)}\right) \times A \quad \text{RPM}$$

where EOD = Exhaust Valve Open Duration (crank degrees);

LEX = Exhaust Pipe Length (inches) from flange to the start of megaphone (racing motor-cycles), taken here to be start of final collector pipe;

LX = Exhaust length (inches), internal to cylinder head from back of valve to flange;

A = Speed of Sound in Exhaust Gases, given by Williams from research by Belilove in 1943 as 18,500 inches/sec.

The various DFV exhaust systems can be estimated from this relation as follows:
for each
EOD = 320^0 and LX = 2.8"

Date	1967	1968	1971	1975
Type	4-2-1	4-1	4-1	4-1
LEX	15" + 15"	30.5"	25.5"	23.5"
Resonant N RPM	7520	7407	8716	9379

Note 84

FORD - COSWORTH DFV

Development 1967 - 1983

Year	1967	1968	1969	1970	1971	1972	1973	1974	1975
DWC or (Principal User) (NoteA)	(Lotus)	Lotus	Matra	Lotus	Tyrrell	Lotus	Tyrrell	McLaren	(McLaren)
CWC or (Principal User)	(Lotus)	Lotus	Matra	Lotus	Tyrrell	Lotus	Lotus	McLaren	(Brabham)
Data Source References DASO	59	543	543	543	42		543	17	17
Typical HP PP (NoteB)	405	415	430	430	450	*450*	460	460	465
@ RPM NP	9000	9500	10000	10000	10000	*10000*	10250	10250	10500
DASO					42				544
Best HP (NoteC)					470				*485*
@ RPM					10000				*10750*
Analyses based on Typical HP									
BMPP Bar (NoteD)	13.48	13.09	12.88	12.88	13.48	13.48	13.44	13.44	13.27
MPSP m/s	19.43	20.51	21.59	21.59	21.59	21.59	22.13	22.13	22.67
MVSP m/s	3.51	3.71	3.90	3.90	3.90	3.90	4.00	4.00	4.10
MGVP m/s	63.5	67.0	70.6	70.6	70.6	70.6	72.3	72.3	74.1
MPDP g	3648	4064	4503	4503	4503	4503	4731	4731	4965
DASO	59								
W kg (NoteE)	162								
PP/W HP/kg	2.5								
DASO	60	60	168	168	544	862	544		544
Price £ (NoteF)	Free	7500	7500	7500	7500	6500	7500		9266
Price @ 2002 Level (NoteG)		79958	75884	71317	65166	52721	55757		47780
2002 Price/PP £/HP		192.7	176.5	165.9	144.8	117.2	121.2		102.8

Year	1976	1977	1978	1979	1980	1981	1982	1983
DWC or (Principal User) (NoteA)	McLaren	(Wolf)	Lotus	(Williams)	Williams	Brabham	Williams	(Williams)
CWC or (Principal User)	(McLaren)	(Lotus)	Lotus	(Williams)	Williams	Williams	(McLaren)	(Williams)
Data Source References DASO	17	543	543	18	18	18	59	59
Typical HP PP (NoteB)	465	465	475	480	485	490	495	510
@ RPM NP	10500	10500	10750	10800	10800	11100	*11100*	11200
DASO	544		19				18 Judd	18 Judd
Best HP (NoteC)	*490*		*495*				515	535
@ RPM	*10750*		*10800*				11300	11300
Analyses based on Typical HP								
BMPP Bar (NoteD)	13.27	13.27	13.24	13.31	13.45	13.22	13.36	13.64
MPSP m/s	22.67	22.67	23.21	23.32	23.32	23.96	23.96	24.18
MVSP m/s	4.10	4.10	4.19	4.21	4.21	4.33	4.33	4.91
MGVP m/s	74.1	74.1	71.6	72.0	72.0	74.0	74.0	74.6
MPDP g	4965	4965	5204	5253	5253	5548	5548	5649
DASO								59
W kg (NoteE)								154
PP/W HP/kg								3.31
DASO	544	544	19				59	
Price £ (NoteF)	10000	12500	17183				26185	
Price @ 2002 Level (NoteG)	44246	47740	60598				56808	
2002 Price/PP £/HP	95.2	102.7	127.6				114.8	

Italics = Approximate Data

Geometric Data

90 DegreeV8 : Bore (B) = 3.373" (85.6742mm): Stroke (S) = 2.550" (64.77mm): B/S = 1.323: Swept Volume (V) = 2987cc
4 Valves per Cylinder @ Included Angle (VIA) = 32 Dergees
Compression Ratio (R) = 11 in 1967 and *assumed* unchanged until 1978 when 12 and then *assumed* same until 1983
Inlet Valve Head Diameter (IVD) : 1967 to 1977 = 1.32" (33.5mm); 1978 to 1983 = 1.36" (34.5mm)
Total Inlet Valve Head Area/ Total Piston Area (IVA/PA) : 1967 to 1977 = 0.306; 1978 to 1983 = 0.324
Inlet Valve Maximum Lift (IVL) : 1967 to 1982 = 0.410" (10.4mm); 1983 = 0.460" (11.7mm)
Inlet Valve Open Duration (IOD) : 1967 to 1978 = 320 Crank Degrees; *assumed* same through 1983
Connecting Rod Length between centres (CRL) = 5.23" (132.8mm): CRL/S = 2.05

Notes

A. DWC = Drivers' World Championship ; CWC = Constructors' World Championship.
B. Not necessarily at the peak of the power curve, which may not have been attainable because of mechanical limits.
Figures are derived mostly from published Team records.
C. From 1971 onwards it appears from refs. (42) and (544) that the best engines gave around 20 HP more than the typical engine.
Ref. (982) reports that there could be 70 HP difference between Best and Worst in mid-'70s, reason unknown.
From 1980 John Judd (Engines Development Ltd.) produced extra power by modifications specially for, and financed by, Williams.
D. **BMPP** = BMEP @ PP and NP.
MPSP = Mean Piston Speed @ NP.
MVSP = Mean Inlet Valve Speed = (IVLmm x NP rpm)/(83.333 x IOD Crank Degrees).
MGVP = Mean Inlet Gas Velocity based on IVA @ NP.
MPDP = Maximum Piston Deceleration @ NP.
E. **W** = Weight (excluding Starter Motor and various Ignition components, mounted on Gearbox).
F. Still free to Lotus in 1968. Includes 8% VAT in 1973 and onwards.
G. Adjusted by General Index of Retail Prices.

Note 85

Peak torque to top-power RPM ratio for DFV

The extent to which the 1967 DFV was limited in RPM for mechanical reasons, short of its 'natural' power-peak speed, can be shown by the following comparisons:-

	1967	1967	1983
Engine	FVA	DFV	DFV
Data Source	583	59	59
$\frac{\text{Peak Torque RPM}}{\text{Top-Power RPM}}$ = F	$\frac{7000}{9000}$ = 77.8%	$\frac{8500}{9000}$ = 94.4%	$\frac{9000}{11200}$ = 80.4%

An F ratio of 80% would have meant a Peak Power RPM (NP) for the 1967 DFV of 8500/0.8 = 10,600.

All these facts were not known in 1967, of course. No power curve was published then for the DFV (nor was any DFV power curve published until one for the 1983 90 mm bore variant (,interim DFY') appeared in 1993 (65), so far as this author knows). However, for whatever reason and whoever supplied it, there was a *misleading* piece of data published at the time of the first DFV announcement in April 1967 - that Peak Torque was 270 lb ft @ 7000 RPM (857). Thus it would then have *appeared* that the F ratio was 7000/9000 = 77.8%, a normal figure.

An experienced observer was *not* misled. 'One of Britain's top engine designers' (probably Harry Mundy, former Chief Designer for Coventry Climax) pointed out that *"if true"* the torque corresponded to 223 psi (15.4 Bar), which he found *"remarkable"* on petrol at R = l1 compared to previous best practice (859). The correct figure for the 1967 DFV, released in 1983, was actually 245 lb ft @ 8500 RPM (59), equal to 203 psi (14.0 Bar). A very fine performance, of course, but not quite so *"remarkable"* as Ford's 1967 Press release would have had its readers believe!

Also in 1967 after the victorious DFV debut another experienced observer, Tony Rudd (then Chief Engineer of BRM), was quoted as estimating that, *if the power curve of the engine were extended to its theoretical peak*, it would give 450 HP (856) as compared to the 405 HP at 9000 RPM which was being quoted by Cosworth.

The new electronic speed governor was not very precise in its limit - a spread of several hundred RPM was quoted for pre-1978 (19) - and it had to be redesigned twice before it was reliable (in 1970 (168) and 1978 (19)). Inadvertently exceeding the intended 9000 RPM mechanical rating may be an explanation for some of the 1967 engine failures.

Note 86

The mid-1971 Tyrrell improvement

The authorised biography of Ken Tyrrell (896, pp159-160) has him ascribing the sudden gain in his car's performance in the 1971 French GP to the addition of an airbox, plus better handling with the first use there of a full-width nose (although saying that the latter did not add much speed). Both these features were copied from the Matra, which had them from the first 1971 race. The shortening of the DFV primary pipes is not mentioned.

This explanation overlooks the fact that the Tyrrells first ran with airboxes in the preceding Dutch GP at Zandvoort. There, the No 1 driver (Stewart) was 0.28% slower on the grid than the Ferrari. As the race was wet and Stewart spun on unsatisfactory tyres and lost ground, while the No 2 (Cevert) had an accident, there was no other data on the airbox gain. Ref (896) puts this as +200 to +300 RPM out of each corner when Cevert was testing at Zandvoort and says he was told not to do a full lap at full throttle so as to conceal the advantage before the next race. Presumably the race practice followed this testing and it is probable Stewart would have used all that the car could give.

At the French race Stewart's grid speed was 0.74% faster than the Ferrari, so 1 % of lap speed had been gained somehow. The candidates for the cause were the DFV improvement and the wide nose, but not the airbox. If 1 % of lap speed was 4% of power at least, about 20 HP, this coincides with the 5" shorter pipe gain on the engine.

An interesting feature of the introduction of airboxes was that F errari also had them at Zandvoort - but the entry was directly behind the driver's helmet - which would have reduced inlet pressure, not improved it! They made no change for the French GP.

Note 87

The new Cosworth Al-alloy casting process

The Cosworth DFX Indycar engine programme began in earnest at the company in early 1977 (351). In service it was found that if there was porosity in the Al-alloy head castings the higher temperature of the TurboCharged engine, running at nearly 2.7 atmospheres absolute intake pressure in 1977, would damage these areas as though they were 'flame-cut'. Keith Duckworth saw this porosity as a failure in consistency of the then-current foundry process and set up a research programme in 1978 under Dr David Campbell to overcome it (869). The patented solution is to use an electro-magnetic (external) pump with no moving parts (developed originally in the French nuclear industry to pump liquid sodium) to force the molten Al-alloy upwards from the middle of an electric-resistance-furnace pool (to eliminate both light and heavy pollutant particles) into a zircon-sand urethane-resin-bonded air-permeable mould (the sand being expensive compared to silicon sand, but capable of great dimensional accuracy, even on small channels, and being recyclable). After this the mould is sealed and inverted to retain pressure until the metal has solidified. The resultant castings are not only much less prone to porosity and, therefore, higher strengths can be relied upon during design but also close-to-size which reduces machining (861).

A new foundry **-** 'precision casting facility' is a better description of something far removed from original 'dirt floor' foundry images! **-** was built at Worcester to use the process. Racing engine blocks and heads were available from it in 1979 (867).

Note 88

The Cosworth DFY and F3000 engines

During 1982 KD was seeking a way to extract more power from 3L NA because he considered that the DFV had reached the most possible with its valve area (850) (although Judd claimed more in 1983, see main text). Variants of the DFV had been built for sports car racing with B increased from 85.674 to 90 mm (type DFL) and this bore was chosen for a Grand Prix redesign with S = 58.8 mm, B/S = 1.531 (an increase of nearly 16% over the original DFV). An increase of IVD from 1.36" (34.5 mm) to 1.42" (36.1 mm) was then possible (IVA +9.0%) (59). Difficulties were experienced immediately with these larger valves - KD said that the cams which produced high RPM did not give the power improvement required and those that gave the power broke valve springs (851). This was to be expected as B/S was increased. The final DFY specification had IVL = 0.432" (11.0 mm) (59), IOD ^ 320° and obtained NP = 11,000 RPM (59) so that MVSP - 4.5 m/s. The 1983 DFV with DA12 cams (IVL = 0.46"), presumably the same IOD, had NP = 11,200 RPM, reaching MVSP = 4.9 m/s.

The short-stroke DFY engine was actually produced in two stages. For early 1983 delivery, a new head retained VIA = 32° over the bigger bore. A batch of 14 units was supplied (McLaren took 6, Ligier 3, Lotus 2 and all these were raced; Williams took 3 but did not race them, preferring their Judd-developed DFVs it seems, according to the engine numbers recorded meticulously in (877) (note that these interim units were labelled by their maker as 'DFY'). The much more extensively redesigned (by Mario Illien) (419) second series DFY appeared in May 1983, 10 units having been bought by Tyrrell at £34,000 each (£70,800 at 2002 level) (544). The VIA was reduced to 22.5° and there was a 27 kg (17%) weight saving from the latest DFV (to 132 kg) - 20 kg saving had been achieved by the first series, partly a consequence of the higher B/S ratio and partly that the crankshaft was altered, as had been done on the DFX, to have only four counter-balances. Another interesting DFX feature used in the DFY was con rod section altered from the conventional I to H. The DFX may have shown that off-centre gas loads on the piston needed this 90° shift of major axis to better resist tilting forces (62, 351, 847); or it may have been that the original US Los Angeles modifiers of the DFV to short-stroke Indy capacity had simply used the local Carillo company, who favoured the H-section anyway (see Note 88B).

As the Ford oval logo appeared on the cam covers for the first time, instead of just 'Ford', it is presumed that they had paid for the development.

Powers claimed for the DFY, both series, at 520 HP were only 10 HP up on the 1983 DFV, although (877) states that the second series had an extra 25 HP between 6500 and 7000 RPM. Nevertheless, the result of all this work must have been a disappointment. The definitive DFY won only one GP, in 1983 on the slow Detroit street circuit, for Tyrrell.

The interim DFY disappeared from GP races during the 1983 season as TC engines were adopted more widely. Tyrrell ran his DFYs through to the middle of 1985, when Renault 1.5L TC units became available. His experience of these engines is interesting as it emerged when he defended himself in 1984 against a charge of breaching the rules in the Detroit GP that year (where second place was obtained and the car post-race inspected) by using water injection 'plus an illegal fuel' into the DFY intakes. Tyrrell denied this and stated the water-only-was simply to improve reliability (by evaporative cooling). He quoted 17 valve-or-piston-related engine failures in 1983 (a mixture of 14 DFV and 16 second series DFY entries, total 30) and none in 1984 up to 2/3 season, using 13 litres of water per race to complement the regulation 220 litres of petrol (544). There was no *direct* power gain, of course, but the possibility of using higher RPM for longer with adequate reliability. The team, nevertheless, were banned from further races and all places and points deducted, because

'infinitesimal' traces of hydrocarbon were found in the water tank.* Tyrrell's evidence does illustrate the knife-edge reliability of even the redesigned Cosworth V8 when giving 500-plus HP ((896) quotes 542 HP for 1984 DFY).

As the door closed on the Cosworth DFV in Grand Prix racing so the Fl Constructors' Association managed to open another for the redundant engine stock - to use them in 1985 and onwards in a new 'stepping stone' formula to replace F2: F3000. Partly to keep costs down but also to limit the power for 'apprentice Fl drivers' moving to F3000 from low-power formulae, the maximum RPM were restricted to 9000 - the original DFV limit - by an electronic governor which cut the ignition for one second if exceeded. Needless to say Cosworth and the rebuilders very soon set about optimising valve timing and porting for higher BMEP at the lower RPM. In later years rival F3000 engines appeared and more development was done on the Cosworth engine, including the use of DFY B and S. The 'original-as-developed' DFV finally began to be phased out of second line racing in 1993, when Cosworth produced a completely new and smaller engine (type AC; see "Significant Other" Fig. SO21A), specifically for F3000, weighing 12 kg (8.5%) less (130 kg v 142) (65). The DFV still appears in Historic racing, as do many of the other engines in this review.

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* Under 1/1000% of the 13L tank, probably because it was filled from the Detroit River. In any case, the 220L petrol tank was only 80% full for that slow race (896).

However, the water-injection system was also a ploy to run an under-minimum 540 kg car. When the tank was topped in a late race stop, 64 kg of lead shot was loaded with the water via a special device! The FISA ruled also that this was illegal ballast because 'unfixed', which Tyrrell disputed because tools were needed to remove the tank (896).

Note 88B

90° shift of major axis on connecting rods

A shift of con rod section from I to H (considered with the crank axis horizontal) had been made in the Wright R3350 radial 18-cylinder 55L aero engine developed for the Boeing B29 Super-Fortress in 1942. It was considered that gas loads off-centre along the gudgeon-pin axis needed a stiffer section to better resist tilting. Previously it had been taken for granted that the rod section priority was to resist centrifugal fling. Clearly the relative level of BMEP against N^z would affect this issue, and the R3350 was highly supercharged.

In 1963 Fred Carrillo founded a firm in California to supply H-section rods to the tuners of US engines. Subsequently they have been, and still are, used in many types of racing engine, eg the TC types Cosworth DFX, Renault and BMW (Eg 64) but later NA engines have I-section rods, eg Ilmor 2175A (see Eg 82) and Ferrari 049 (Eg 85).

Fig. N88B(A)
Cosworth DFV
I-section con.-rod.
DASO 858 Motor 6 May 1967

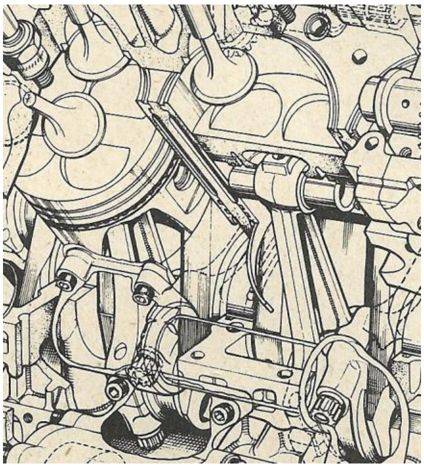
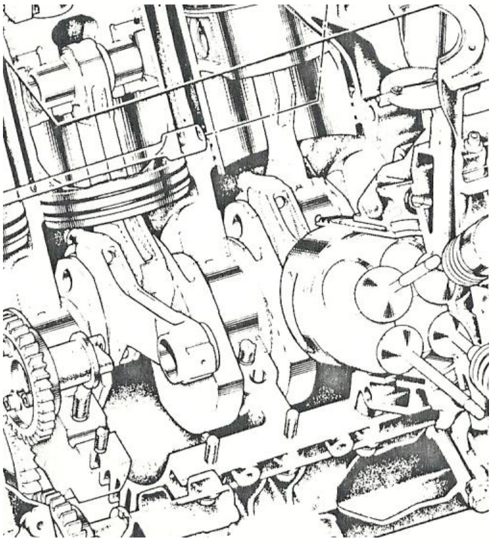


Fig. N88B(B)
Cosworth DFX
H-section con.-rod.
DASO 59



Note 89 TurboCharging background

[This note is part of web site http://www.grandprixengines.co.uk]

Origin

The idea that the exhaust gas of an internal combustion piston engine could be led through a turbine which would use the otherwise-wasted energy to drive an inlet-charge compressor is as old as the Grand Prix car. The Swiss Alfred Buchi patented the basic separate turbocharger (TC) principle in 1905 (569,685).

Fundamental relation

The fundamental relation governing TC is:-

$$\left(1 - \frac{1}{RT^{1/4}}\right) = \left(\frac{(IVP)^{1/3.5} - 1}{\left(\frac{EGT}{250}\right) \times \left(\frac{MT}{MC}\right) \times eo}\right)$$

Where:- IVP = Inlet-charge absolute pressure ratio above ambient, atmospheres absolute (ATA);
RT = Turbine absolute expansion ratio to ambient necessary to drive the Compressor;
EGT = piston engine Exhaust Gas absolute Temperature at Turbine inlet;
MT = mass flow rate through the exhaust Turbine;
MC = mass flow rate through the Compressor;
MT is greater than MC by the fuel flow rate where the fuel is fed into the charge after the compressor. [It is assumed that any turbine wastegate is closed.]
eo = Overall TC Efficiency = (Mechanical x Turbine x Compressor Efficiencies).

This relation is derived in Sub-Note A.

For automotive use the fundamental problems of the turbocharger were :-

- To obtain a sufficiently high value of 'eo' at the necessarily small size of the Turbine and Compressor;
- To obtain Turbine materials which could withstand 'EGT' around 1,300K from a petrol engine for a sufficiently long life.

Early aero-engine research

In WW1 and the early 1920s the RAF/RAE did research into TC to restore aero-engine power at altitude. James Ellor of that establishment post-WW1 did the aerodynamic design of units to suit, firstly, the Napier "Lion" and then the Rolls-Royce "Condor". In 1924 the Mark V of the latter type was built with an exhaust turbine driving a 2-stage centrifugal compressor and air-air intercooling before the engine intake , the fuel being petrol of low evaporative cooling capability. However, either the turbine flow capacity was insufficient or the component efficiencies were too low – both, probably – so that the back pressure ahead of the turbine actually reduced the ground-level power compared to the naturally-aspirated build. No flight testing was done and the TC project was abandoned (900,901). British piston aero-engines from then onward relied on mechanical supercharging (MSC).

Early racing-car TC

In 1925 Major Frank Halford, closely associated with both the aero-engine and racing-car scenes, designed from scratch for a racing car a 1.5 Litre IL6 engine with TC. Presumably the engine was intended to compete under the 1926-1927 1.5 L Grand Prix formula. This Halford engine had a centrifugal compressor with intercooler (although the fuel would certainly have been alcohol-based with high evaporative cooling capability; Halford, with Harry Ricardo, had been a pioneer of that type of fuel for naturally-aspirated racing engines a few years earlier) driven by an axial turbine (286). The blower on test was inefficient (904) and the turbine (with very short unshrouded blades) would certainly have been so. See P.7 for a section of the engine. The engine never raced as designed and it was converted to Roots MSC.

Nothing further was heard of TC in the motor-racing world until 1952.

Successful aero-engine TC by General Electric (GE)

Led by Dr Sanford Moss, GE of the USA eventually developed by the late 1930s a TC unit suitable for altitude power boosting of military aero-engines built by Wright and Pratt & Whitney. Dr Moss had worked on centrifugal compressor research since before WW1 and in the 1920s was helpful to Duesenberg and Miller as these firms developed their US track-racing engines using that MSC approach (6). It seems very likely that the "*can-do*, *cut and try*" methods of the racing men also benefitted the GE work. The GE turbocharger found its 1st successful application with the Wright R1820-51 engine fitted to the prototype Boeing Y1B-17A in 1938. Compared to the MSC R1829-39 in the preceding service-trials B-17 batch this raised the operating ceiling by 10,000 feet (to 25,000 ft) (902) and so started the useful career of that aircraft as a day bomber. Subsequently GE applied their TC units to other US aero-engines which saw wide service in WW2 (see Fig. N89D on P.7). The basic GE design was an axial turbine driving a centrifugal compressor, (but with a size advantage on efficiencies compare to Halford's car TC). A large step forward was made in reliability by the use of "Vitallium" (Haynes Stellite 21) turbine blades (composition 62% Co, 28% Cr, 6% Mo, 2% Ni, 2% Fe + 0.2% C) which retained useful strength to 1,000C exhaust temperature (685). This GE TC experience led to the firm's selection by the USAAF to receive and develop the Whittle turbojet engine given to the USA under "Reverse Lend-Lease" in 1941.

Post-WW2 Diesel truck engine TC

The Diesel engine is a natural type to be TurboCharged since the exhaust is some 400C cooler than a petrol engine so turbine materials can be cheaper (although, of course, the heat energy to be exploited is also less).

By the early 1950s the TC design had been improved greatly for commercial Diesel engines by applying a radial-inflow turbine, more efficient at automotive sizes than the axial type, to drive the centrifugal compressor. Also the unit was sized to provide the desired boost pressure at medium engine speeds with an added wastegate to bypass surplus exhaust gas around the turbine at higher speeds so as to hold that pressure constant.

TC truck engine racing at Indianapolis

In 1950 the US Cummins truck engine company had entered unsuccessfully for the Indianapolis 500 mile race with a Roots MSC special racing version of their standard Diesel engine, taking up a 6.6 lItres option given for Diesels in the regulations where the pure racing spark-ignition engines were limited then to the alternative of 4.5 litres Naturally-Aspirated (NA) or 3 litres Pressure- Charged (PC).

Cummins re-entered the '500'race in 1952 with a TC Diesel engine. This provided about 400HP where the MSC engine had given 320HP, both at about 2 ATA IVP, i.e. + 25% (12,569). Despite the car being 1,000 lb heavier than its alcohol-fuelled rivals this power was sufficient (the engine horizontally mounted in a racing chassis) to take the Pole at 138 mph*. As the unit did not use the 'oversized TC + wastegate system' the basic centrifugal compressor relation of

$$\text{Pressure Rise proportional to } (\text{RPM})^2$$

meant that, even with the relatively small RPM drop in backing-off the throttle for each of the 4 large-radii slightly-banked Indianapolis corners, there was a substantial lag before full boost and power returned for the straights until the driver anticipated it and re-opened the throttle "early".

However, in the race the compressor suffered dirt ingestion clogging from the low intake of the flat engine and retired at 36% distance (569,905).

This was the 1st time that a TC engine had appeared in motor-racing and it would be the last for another 14 years.

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*At Indianapolis the Pole is taken by the fastest car on the 1st day of official 4-lap Qualification, regardless of later higher speeds. Subsequently a 4.5 L NA car went 0.15% faster and then a 3 L MSC car went a record 0.74% faster. The rule-setters must have congratulated themselves on a job well done for three such completely different types of engine!

Successful Indianapolis TC

After 1946 the NA engine had reigned at Indianapolis in the shape of the Meyer-Drake Offenhauser (originally Miller) IL4 (of 4.5 L, reduced by the rules to 4.2 L in 1957).This was finally displaced after 18 straight wins by the Ford 4.2 L V8 NA in 1965. Louie Meyer then left the partnership and became the agent for the Ford engine. Dale Drake then asked Leo Goossen (the long-time sole American racing-engine designer) to redesign the NA 'Offy' very substantially into a short-stroke 2.8 L PC (the then alternative regulation pure-racing size).

A Roots MSC system was chosen by the maker for 1966 (716) but another solution was offered by Garrett AiResearch, using a commercial truck TC with wastegate (569,711). This contrasting pair therefore provided a very good illustration of the merits of TC v. MSC, as detailed below.

Data sources	(569,711,716,906,907,1046)	
Engine maker	Drake	
Bore (B)/Stroke (S)	4 1/8'' (104.775mm) / 3 1/8'' (79.375mm) = 1.32	
Swept Volume (V)	167.05 cubic inches (2,737 cc)	
	Mechanically Supercharged (MSC)	TurboCharged (TC)
Compression ratio (R)	8	8
Fuel	Methanol	
Pressure charging (PC) system		
	Commercial Roots type, made by Meihler-Dexter	Commercial Garrett AirResearch Type TE06
Fuel supply	2 Hilborn injectors at MSC entry	Hilborn injectors to each port, downstream of TC
Absolute pressure at inlet valve (IVP)		
	17 psi boost = 2.16 ATA Rising from 15 psi at 6,000 RPM	17 psi boost = 2.16 ATA limited by exhaust wastegate from 5,000 RPM
Intercooling	(Not needed with Methanol fuel)	
Peak Power (PP) HP	530	626*
@ NP RPM	8,500	8,500
Weight (W) kg	172	163
PP/W HP/kg	3.08	3.84

Conclusion: with all other things being equal, the TC engine therefore gave 18% more power than the MSC while being 5% lighter, a Power/Weight advantage of 25%.

The 96 extra HP of the former came from the shaft HP extracted from the crank to drive the MSC supercharger being that much greater than the power deducted pneumatically from the TC engine on the exhaust stroke by the increased back pressure (above the free exhaust expansion when NA) needed to drive the turbine.

The Roots supercharger required a gross 60HP @ 6,000 engine RPM and 15 psi boost (716) which can be extrapolated to at least 60 x (8,500/6,000) x (17/15) = gross 96 HP at full speed (642). Although this 'envelope back' calculation implies *zero* loss from the TC back pressure increase, which cannot be correct, it does illustrate how the TC advantage arises*.

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*626 HP quoted from (711). Ref. (1046), equally authoritative, has 600 HP for the TC engine, i.e. 70 HP (13%) advantage. This accords better with the calculated 96 HP supercharger gross driving power, on the basis that 96 – (a deduced 26 HP extracted from IHP by back pressure in the TC engine) = 70 HP.

The difference of about 4% in quoted TC power, which may be just manufacturing repeatability, illustrates the problem of historical analysis described in the Foreword!

Further development of the TC engine at Indianapolis

Neither type of Drake engine could hold its power for long in 1966, but the TC version was clearly the '*Way to go*' and so it received more development for 1967 although still not a winner. However, success came in 1968. The TC innovation had prompted Ford that year also to produce a short-stroke 2.8 L adaptation of its original 4.2 L NA V8. From 1968 to 1996, after which date NA was imposed by Indianapolis rules, TC was supreme at that track whatever the engine maker (Cosworth and then Ilmor competed later) or the swept volume specified or the maximum IVP allowed.

Special conditions for TC at Indianapolis

Because the rules required methanol fuel (as a safety feature, it having a higher flash point than petrol (gasoline)), no Indy engine needed an intercooler because the high evaporative heat value of that fuel reduced the charge temperature entering the cylinder from the compressor delivery figure.

Indy racing, on a track with 4 large radii, slightly-banked corners, did not demand much attention to the lag in boost pressure recovery from the centrifugal compressor if the throttle was shut for any significant time, as would occur on European road circuits. Consequently only 1 turbocharger was fitted to Indy engines since its inertia was not a particular problem.

Sports-car TC

After 1968 TC was applied to several types of Sports-racing and Touring-racing engines by BMW, Porsche and Renault (all of whom went on to build TC Grand Prix engines). These engines ran on petrol (although Porsche did experiment with Toluene-based fuel once (608)). Despite this and having no intercooler and also being air-cooled, the Porsche Type 917/30 F12 produced 1,100 HP from 5.4 L with IVP = 2.3 ATA in 1973 to power the CanAm sports-racing Champion, repeating the previous year's success with a 5 L TC engine. This 1973 figure was then a record output for a road-racing car. These Porsche engines had coped with throttle lag by using small low-inertia turbochargers to each 6 cylinder bank and boost-sensitive fuel injection (569). Probably detonation was prevented by using a very-rich mixture. Certainly, when a fuel ration was imposed by rule in 1974, Porsche withdrew from the CanAm races.

Renault Grand Prix TC

In 1973 Renault produced in their "Amadée Gordini" specialist plant a pure-racing 90V6 2 L engine, NA, Bore (B)/ Stroke (S) of 86mm/57.3 = 1.5 for sports-car competition. This was Type CH1, designed under the direction of Francois Castaing (the 'CH' designation was in memory of Claude Hard, a former technical manager). After development this powered the winner of the 1974 European 2 Litre Sports-car Championship. The engine also powered the F2 Elf and Martini cars which won European F2 Championships, the former make in 1976 and the latter in 1977.

In 1975 a TC version was built and raced at Le Mans the following year but DNF. This had an intercooler to suit the petrol fuel. With Ken Tyrrell's advice (896) Renault had also since early 1975 been evaluating the 1.5 L PC rule option in the Grand Prix field, initially with a reduced-Bore/ short-Stroke (80mm/49.4 = 1.62) TC version of the CH1. While preparing in late 1976 for the 1977 Le Mans Renault senior management decided also to enter Grand Prix racing the same year. This dual-effort instruction showed the defect of "Big Firm Management" – an inability of 'mahogany-office-dwellers' to understand the load imposed by their decisions on 'coal-face-workers'! Le Mans was *not* won in 1977 but the GP car with Type EF1 engine *did* appear in July at the British race – a DNF. This was the 1st TC GP car since Halford's abortive design of 1925. In this case the initials 'EF' recognised financial help from Elf, the French national fuel company

Illustrations of the 1978 Renault EF1 are given on P.5 (see Figs. N89A & B.)

Development of both the 2 L Le Mans and 1.5 L Grand Prix engines and cars continued in 1978 and this time a Le Mans victory was achieved. Afterwards all effort was put into the GP machines (909,910). A 1st win was gained in 1979, very fortuitously at the French GP, shortly after changing from a single TC to dual TC, one per bank of lower inertia, (amongst a vast number of reliability improvements).

The simple *destroking* of the CH1 decided upon finally for the 1977 Type EF1 GP 1.5 L engine (485) had led to the extreme B/S of 86mm/42.8 = 2.01, a ratio never seen before and which limited engine speed by reaching a Mean Valve Speed (MVS) limit for the steel coil-spring return system (CVRS) long before any bottom-end limit. Eventually, Renault accepted this and, in 1985, they revised the dimensions back to the original 80.1mm/49.4 = 1.62 for the type EF15.

The EF1 was the 1st Grand Prix PC engine to take a well-developed NA design having individually-tuned inlet and exhaust systems, a principle which had entered the GP arena originally in 1952 with the Ferrari Type 500 (q.v. Eg. 30), and simply increase the Manifold Density Ratio (MDR) by enclosing the inlet bells in a plenum chamber fed by the TurboCharger. The inlet tuning and, as far as possible to the TC entry, the exhaust tuning were preserved.

Bernard Dudot was the principal development engineer throughout the Renault unit's TC life, assisted by Jean-Pierre Boudy and later Jean-Jacques His. It was Boudy, stimulated by the valve-gear problem of the EF1, who invented in 1984 the "Distribution Pneumatique" system with gas springs in place of steel springs which became the standard for Grand Prix engines after 1990 and permitted B/S ratio to be pushed well over 2 in the following decade (see Eg. 73 et seq). Honda later named it "Pneumatic Valve-Return System" (PVRS), which is more descriptive of its function.

Having pioneered TC in Grand Prix racing Renault saw itself copied in that sphere by all other racing engine makers, beginning with Ferrari in 1980. They never won either the Drivers' or Constructors' World Championships with their own TC cars (over 1977 – 1985) or as TC engine suppliers to other chassis makers (Ligier and Lotus in 1984 – 1986, plus Tyrrell 1985 – 1986). Partly this was because of another bad "Big Firm Management" decision which was to fire their No. 1 driver Alain Prost at the end of 1983, after he was just beaten by Piquet in a Brabham-BMW to the Drivers' title, because he dared to criticise them – thereby losing a driver capable subsequently of winning 4 Championships!

1978 Renault EF1

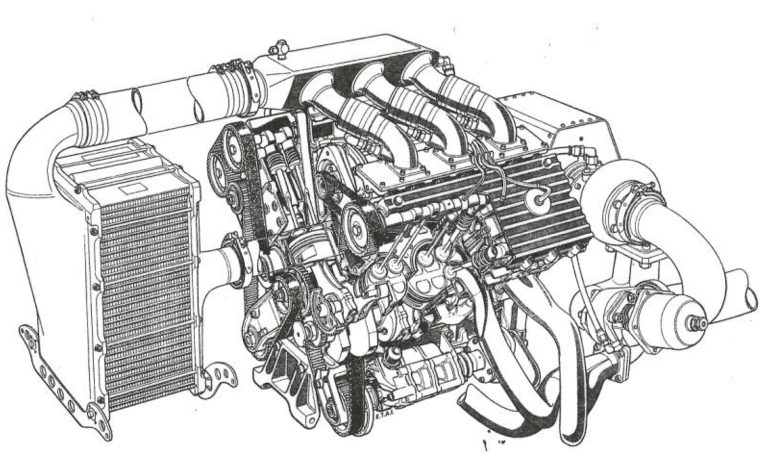


Fig. N89A

90V6 86/42.8 = 2.009 1,492 cc
4 valves per cylinder.
Belt drive to camshafts.
Intercooler required with 102RON petrol to reduce inlet charge temperature.
Plenum chamber feeding tuned inlet tracts.

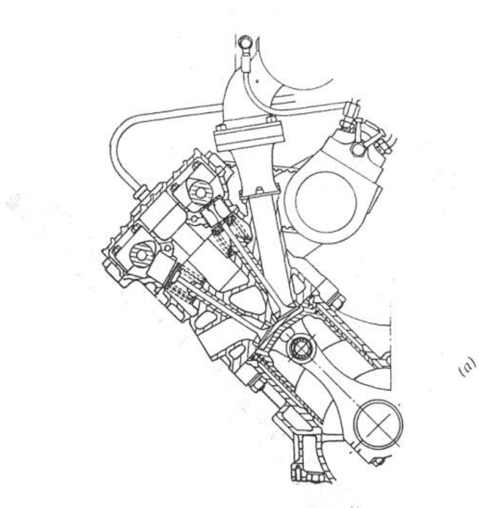


Fig. N89B

VIA = 21.5^0

This section is drawn for the dual turbochargers fitted after mid-1979 with separate plenum chambers for each bank of cylinders.

Note 89

Sub-Note A

Derivation of fundamental TC relation

The power developed by a Turbine / required by a Compressor is:-

$$Cp \times \triangle T \times M$$

where:- Cp = Specific Heat of air at constant pressure appropriate to the Turbine / /Compressor temperatures;

$\triangle$T = Total Head Temperature drop / rise of air passing through the machine;

M = Mass Flow rate through the machine component.

At equilibrium speed, where the Turbine supplies, with some rotor/bearing loss, just the power required by the Compressor:-

(1) $$em \times CpT \times \triangle TT \times MT = CpC \times \triangle TC \times MC$$

where T = Turbine; C = Compressor; and 'em' = Mechanical Efficiency of the rotor system.

Basic thermodynamic theory (e.g. (911)) also provides:-

(2) $$\triangle TT = et \times EGT \times \left(1 - \frac{1}{RT^{1/4}}\right)$$

(3) $$\triangle TC = \left(\frac{T1 \times (RC^{1/3.5} - 1)}{ec}\right)$$

Where 'et' and 'ec' are Turbine / Compressor Efficiencies;

RT = Turbine absolute total head expansion ratio to ambient necessary to drive the Compressor;

RC = Compressor absolute total head pressure ratio; if a small loss of pressure through an intercooler is neglected then RC = IVP = absolute Pressure ratio at the Inlet Valve above ambient;

EGT = Exhaust Gas absolute total-head Temperature at Turbine inlet;

T1 = ambient absolute total-head Temperature at Compressor inlet.

Combining equations (1), (2) and (3) :-

(4) $$\left(1 - \frac{1}{RT^{1/4}}\right) = \left(\frac{(IVP)^{1/3.5} - 1}{\left(\frac{EGT}{T1}\right) \times \left(\frac{MT}{MC}\right) \times \left(\frac{CpT}{CpC}\right) \times em \times et \times ec}\right)$$

It is convenient to put

$$eo = (em \times et \times ec).$$

At the usual temperatures:- CpT = 0.276; CpC = 0.24;

and T1 = 288K (Standard ambient 15C).

Therefore (4) can be reduced to:-

(5) $$\left(1 - \frac{1}{RT^{1/4}}\right) = \left(\frac{(IVP)^{1/3.5} - 1}{\left(\frac{EGT}{250}\right) \times \left(\frac{MT}{MC}\right) \times eo}\right)$$

Fig.N89C
1925 Halford Special
IL6 63/80 = 0.789 1,496 *cc*
Designed by Major Frank Halford.
DASO 286 p.93

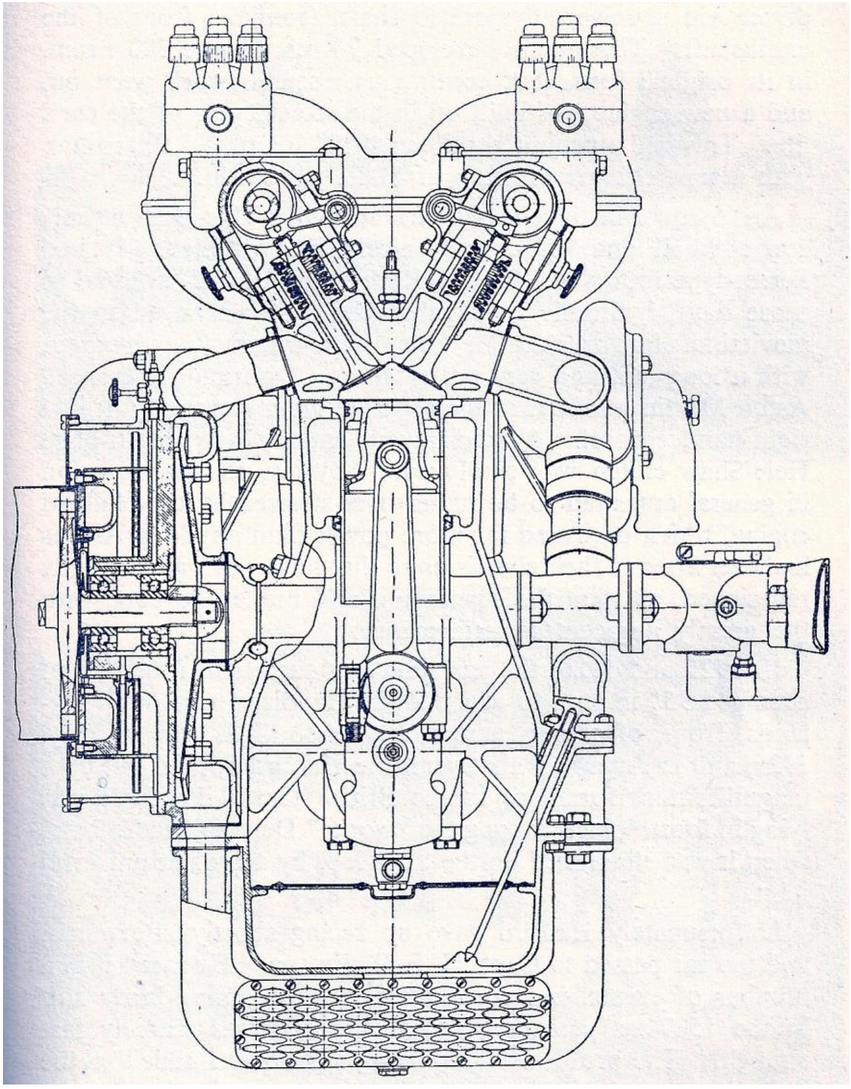
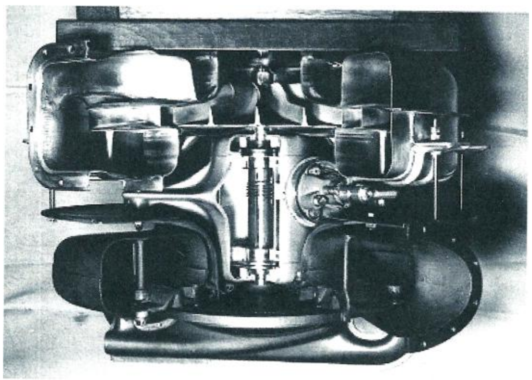


Fig. N89D
1942 General Electric TurboCharger
The exhaust system incorporated a wastegate to control turbine speed and therefore boost.
The unit could be installed either way up; this shows the axial turbine rotor at the bottom.



Adding a GE TurboCharger to the MSC Pratt &Whitney R-2800 radial 18 cylinder 46L aero engine increased the Thermal Efficiency from 28% to 33%.
DASO [US] A*ir Force Magazine* Sept. 1982, quoting a NACA report.

Note 90

Knock-resistance in Pressure-Charged engines

(and some description of Naturally Aspirated special fuels 1989 – 1992 in Sub-Note B)

Note 58-2 describes how pre-WW2 tests with super-rich mixture in supercharged aero engines first enabled more power to be produced than the standard Naturally Aspirated (NA) variable compression fuel-rating engine would predict, by raising knock-resistance. Possibly a memory of this surfaced 40 years later, after the introduction by Renault to Grand Prix racing of the TurboCharged (TC) engine in 1977. This was 1.5 L under the Pressure Charged formula which was alternative to the NA 3 L (but never used by any firm previously), burning 102RON petrol in accordance with the current regulations and therefore needing an intercooler before inlet valve entry to reduce the temperature from the compressor.

Very-rich petrol mixture for TC

Initially very-rich petrol fuel/air mixture was employed by Renault, as in the WW2 aero engines. By 1981 their pioneering 90^0V6 (with an Inlet Valve Pressure (IVP) of about 2.7 Bar and a Compression Ratio (R) of 7 (571,909)) was consuming 250 litres in a 300 km race with a maximum of 550HP (574). The rival 3 L Cosworth DFV units had 10% less power but needed only 170 litres (544), i.e. the TC engine was running 34% richer than the NA ([250/550]/[170/500] = 1.34). The latter engine was almost certainly running for maximum power at 20% richer than Stoichiometric Fuel/Air Ratio (SAFR), i.e. at AFR = 12.2 so that the TC would have been around AFR = 9.2.

Ferrari soon followed Renault's lead on TC (CoY Egs.63, 65) and then Brabham fitted a TC BMW engine.

For all these engines the racing need was to reduce fuel consumption and a fuel was required which would pass the conventional regulation fuel test but produce higher-than-expected power at high Manifold Density Ratio (MDR).

Toluene

The well-known high knock-resistance of Toluene was investigated again. Harry Ricardo had tested this as far back as 1919 and had then proposed it, blended with varying %ages of normal-Heptane of zero knock-resistance, to form a scale to rate commercial fuels in his pioneering E35 variable-compression engine (242) (this basic idea was reused in 1927 by Edgar with the substitution of iso-Octane for Toluene and this was then adopted generally (592)).

A special fuel with equal parts of Toluene, petrol and ethyl alcohol (plus TEL) was supplied by BP for the Rolls-Royce 'R'-type engines built specially for the Miss England II and III Water Speed Record-contending motor-boats of 1930, 1931 and 1932 (52).

In WW2 Toluene was rated at rich mixture as "Over 160" on the Performance Number scale but it was not used in service fuels as it was of more value in the explosive Trinitrotoluene (TNT) (599).

Use of Toluene in motor-racing

Apparently the next use of Toluene was in an ostensibly "petrol" regulation-meeting fuel used by Porsche in their type 936 sports cars with TC F6 2.1 L engines from 1976 (608).

At any rate, for Grand Prix use initially by BMW from mid-1983 in their IL4 1.5 L TC unit Toluene was used by the German firm of BASF-Winterschall with the original Ricardo principle of blending just sufficient n-Heptane to drag it down in the conventional test to the regulation 102RON. With the higher Inlet Valve Pressure (IVP) then possible this brought Brabham with BMW engines the 1st TurboCharged Drivers' Championship (CoY Eg. 64), defeating the TC-pioneering Renault team still using "real" petrol (although with water-spray cooled intercoolers (569)).

Honda fuel details

The other teams with TC engines – which soon included Porsche (aka TAG (CoY Egs. 66, 67, 68)) and Honda (Egs. 69, 70, 71) – had to follow BMW's lead with Toluene-base fuel* for the next 5 years, 1984 – 1988, until Pressure Charging was banned once again after a programmed taper-down of TC power by limits on IVP and a new race-fuel-ration rule: 4 Bar & 195 L in 1987 and 2.5 Bar & 150 L in 1988. In 1986, before these limits came into force, Honda's 1.5 L RA166E (Eg. 68) reached IVP = 5 Bar on R about 7 and produced 1,200HP for qualification, i.e. on its Toluene-base fuel with IVP about doubled from the petrol-burning 1981 Renault it had about double the power.

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* It is amusing that the slang term for these highly-knock-resistant products was "Rocket fuel" since the bulk of liquid-fuelled rocket launches up to that time had been burning *kerosene*, a very poor resister to knock (maybe 20 Octane rating)!

As a well-reported engine (20, 535), the 1988 80^0V6 1.5 L TC Honda (Eg. 71) at the reduced-by-regulation IVP of 2.5 Bar, but with R raised to 9.4 to take advantage of the fuel to reduce fuel consumption as needed with the 150 L ration, used an 84% Toluene+16% n-Heptane fuel run 15% rich (AFR = 11.9) for Qualification and at 676HP produced 10% more power than “real” petrol (presumably run super-rich). How this fuel remained under the 102RON rule limit in the low-speed calibration engine is illustrated in Sub-Note A.

Power v. Consumption

TurboCharging led easily to the use of much higher power for Qualification at very short engine life and excessive fuel consumption in order to gain the high starting grid position which had become even more important post-1967 because the introduction of aero downforce, with its upward flowing wake, made passing more difficult. An engine change to a new unit with different settings would then be made for the race.

As a fairly extreme example, the 1988 Honda RA168E (mentioned above) on the same fuel in race tune, while retaining the regulation IVP 2.5 Bar, had the intercooling restricted to provide 70C engine inlet temperature instead of 40C, AFR increased to 13.4 instead of 11.9 and fuel pre-heated to 80C. These changes dropped power by about 10% but reduced power-specific fuel consumption by about 12% and enabled race consumption to be kept within the 1988 regulation 150 litres at winning speed.

In-race adjustments

In the TC era, apart from their throttles which tended to be either wide open during acceleration (64%) or shut while braking (20%), with 16% of juggling in corners (20), all drivers were provided eventually for the race with several power settings by push-button adjustment of the electronic Engine Management System, from “Max. Power” for overtaking to “Min. Consumption” as described above, relying on their attention to a sophisticated fuel gauge to ensure that they finished. There were further electronic aids by data telemetered to the pit staff and 2-way radio to permit pit managers to control their employees (theoretically!). Direct engine resetting by radio from the pit would have been possible but was banned.

Continued on page 3 Sub-Note A

Sub-Note A

Toluene-base fuel in Calibration and in Racing engines

There is an interesting comparison between the 84% Toluene+16% normal-Heptane fuel used by Honda in 1988 in the RA168E 80^0V6 TurboCharged 1.5 L (running at optimum 15% rich) and the blends of those constituents tested by Ricardo in 1920 in his E35 Naturally.Aspirated variable-compression 2.1 L research engine (1 cylinder Bore (B) 4½''/Stroke (S) 8'' = 0.563), as described in (242).

Although those early tests were limited to 79% Toluene+21% n-Heptane and reached Compression Ratio (R) of 7.5 a reasonable extrapolation of the data shows that an 84/16 mix would have reached R = 9. This compares with the RA168E operating at R = 9.4 ***on top of an Inlet Valve Pressure (IVP) of 2.47 Atmospheres Absolute (ATA)*** (2.5 Bar rule). The reason for the far superior knock resistance of the Japanese engine must lie in the following differences:-

	E35	RA168E
B/S	4½''/8'' = 0.563 (114.3mm/203.2)	79mm/50.8 = 1.555
1 cylinder volume cc	2,085	249.0
Manifold Density Ratio (MDR)	1	2.27 (2.5 Bar at 40C)
RPM	1,500	12,500
Mean Piston Speed (MPS) m/s	10.16	21.17
Inlet charge Mean Gas Velocity (MGV) m/s	25.71	68.73
$\left(\frac{\text{Squish Lands}}{\text{Piston Area}}\right)$	-23.5% [i.e**. Negative** because the Combustion Chamber was 5'' dia. over 4.5'' bore]	15%
Flame Travel mm	57.15 2 x Side plugs	39.5 1 x Central plug
Combustion Chamber $\left(\frac{\text{Surface Area}}{\text{Volume}}\right)$ cm^{-1}	1.3 @ R = 9	≈ 3.*3* @ R = 9.4 (assuming squish lands were not part of the chamber)

Most of the factors listed were favourable to the RA168E in resistance to knock although it is impossible to apportion shares.

The above comparison illustrates the way in which a fuel rated in the standard calibration engine, which was similar in principle to Ricardo's E35 (the CFR 1 cylinder 3¼''/4½'' = 0.722) could nevertheless give far superior performance in a modern Pressure Charged racing engine

Continued on page 4 Sub-Note B

Sub-Note B

Special fuels in Naturally Aspirated racing engines, 1989 –mid 1992

Following on from their success in outflanking the 102RON petrol limit in TC engines the fuel suppliers in the ensuing 3.5 L NA era produced special blends for the same purpose. It seems that these also produced extra power by permitting higher compression pressure than could be used without knocking on “real petrol”, to judge by the change found necessary by Renault (burning Elf fuel) in mid-1992 when the fuel rule was tightened to exclude *“Power-Boosting-Additives*”. According to (919) this required revised ignition settings in the RS4 engine which, in the context of reduced power, could only mean a *smaller* ignition advance before TDC. This *may* have been a temporary expedient until lower compression pistons could be obtained.

In the Honda RA122E/B engine at the same date the fuel change resulted in a power drop of 5% (816PS to 775PS)(69). On the basis of Air Standard Efficiency this drop would occur if Compression Ratio (R) was dropped from the stated 12.9 to 10.5. There may have been some offsetting gain of Mechanical Efficiency which would have reduced the drop of R required and there may have been some alternative retardation of the ignition, as in the Renault. Expressing the power drop only as reduced R looks unlikely and it is possible that the special fuel (Shell in the Honda case (535)) *had* beaten the “Tizard-Pye law” (see Note 10) by a small amount.

Fuel 2000 – 2009

In the series of BMW Grand Prix engines over 2000 -2009, ref (1095) states that, despite the tighter fuel regulations, it was still possible over that period by fuel development to obtain a 1% performance increase and a 2% reduction in consumption.

Note 91

BMW fuel from mid-1983

As well as the advantage described in Note 90 for Toluene-base fuels over “Real Petrol”, the BMW fuel supplier according to (928) made use of an imprecision in the *tolerance* permitted in the rules to provide them with fuel which post-race calibration engine analyses showed as between 102.5 and 102.9RON, where the rule was nominally 102. Renault and Ferrari asked for a clarification at the end of the season, although not actually protesting against the results, and eventually (October 1984) the figure of 102RON was specified as “Not-to-be-exceeded”.

That level had been set in the first place as “5 Star pump petrol of the day at 101RON +1 tolerance”. Alain Prost (Renault, who had lost the 1983 Championship by 2 points), had wished his team to protest during the season which could have led to the 1983 Champion, Piquet, being disqualified but Renault did not wish to win by an off-track committee decision (928).

Of course, tolerances are given in any technical specification to allow batch production at a reasonable cost. They are supposed to be Plus or Minus so that the *average is equal to the nominal design value* – **not** at the edge of the tolerance band. Renault pointed this out (928).

However, in a competitive situation the temptation to push to the limit is always present!

Note 92
Honda: Racing Motor-Cycles

		All NA	
Year	1966	1967	1984
Type	RC149	RC181	NR500
Data Sources	14,228,354	14,228,354 357	16,74,228, 354,357,358
Fuel	All 100 Octane petrol		
Configuration	IL5	IL4	90V4
Valves per Cylinder	4v/c	4v/c	8v/c
@ Included Angle (VIA)	56^0	75^0	40^0
Bore/Stroke (B/S)	35.5mm/25.1 = 1.414	59.4mm/44.9 = 1.323	"75.36"mm/28 = "2.691" Note (1)
Swept Volume (V) cc	124	498	499
Compression Ratio (R)	10.7	11.5	12.1
Peak Power (PP) HP	32.7 Note (2)	95.2 Note (2)	134 Note (3)
@RPM (NP)	20.000	14,500	19,500
PP/V HP/L	263	191	268
Brake Mean Effective Pressure @ NP (BMPP) Bar	11.78	11.80	12.31
@ Mean Piston Speed (MPSP) m/s	16.73	21.70	18.20
Mean Gas Velocity at Inlet @PP (MGVP) m/s	58	66	68
	All Coil-spring Valve Return System (CVRS)		
Mean Valve Speed @ PP (MVSP) m/s	2.9	3.7	4.0
BNP m/s	11.8	14.4	"24.5" Note (1)

Note (1):- Equivalent circular Bore with same piston area as "Oval" cylinder.

Note (2):- With exhaust diffusers ("megaphones").

Note (3):- Racing motor-cycles had to have exhaust silencers by this date but the quoted test results may have been un-silenced, although with plain pipes, not megaphones.

Note 93

Honda engine designation system

From the start of their Formula 2 programme in 1978 until departure from Formula 1 at the end of 1992 Honda racing engines were designated according to the following new system:-

RA (3 figures) **E** = **Racing Automobile** (.........) **Engine**
(In Japan, because of the nature of their own pictographic written language, English is (or was) often used in scientific or engineering reports*);

The 3 figures were selected as:-

1st **Formula** i.e. **2** or **1;**

2nd **No. of cylinders.** When F1 10 or 12 cylinder engines were built, the 1st and 2nd figures were combined (possibly because printed drawing forms only had 6 spaces for type designation!);

3rd **Last figure of the year**.

This designation system had to be extended in the 3NA era as competition drove many modifications during the season. The basic year code was given a "Version x" to identify a change. When the 1992 V12 was redesigned completely it was designated RA122E/B (69).

*When, after the end of the 2nd Pressure-Charged era, Honda were prepared to release data on their RA168E TurboCharged engine, they chose to write a paper in English for the American Society of Automotive Engineers (SAE), (DASO 20) (see Appendix 3).
After they departed from Formula 1 in 1992 they were once again prepared to give details of their last 3rd Naturally-Aspirated era engine, the RA122E/B. This time they gave the paper in Japanese to the Japanese Society of Automotive Engineers (JSAE) (DASO 69). This was translated by Weslake Developments and a copy was supplied to the author by courtesy of their Managing Director, the late Brian Lovell. Perhaps using their own language showed the confidence of Honda after a 6 year series of powering Champion cars.

Note 94

Grand Prix 1.5L TC engines developed from F2 2L NA

Some interesting comparisons between Normally Aspirated (NA) and TurboCharged (TC) engines can be made from the 1977-1988 developments by Renault, BMW and Honda of their F2 2L NA engines into Grand Prix 1.5L TC units, in each case the basic changes being to shorten the Stroke and lower the Compression Ratio (apart from adding the TurboCharger, of course, feeding into a plenum chamber from which individual and tuned inlet tracts fed the cylinders as in the NA application).

Data are tabled below. In each case the F2 engine is the fully-developed specification and the TC is the initial version.

All engines, NA or TC, running on 102RON "real" petrol

F2 NA i.e. Manifold Density Ratio (MDR) = 1

Year	1977	1982	1983
Make	Renault	BMW	Honda
Type	CH1	M12/7	RA263
Data Sources	910	454	680,929,931
Configuration	90V6	IL4	80V6
Valves per Cylinder	4v/c	4v/c	4v/c
@ Included Angle (VIA)	21.5^0	40^0	40^0
Bore/Stroke (B/S)	86mm/57.3	89.2mm/80	90mm/52.3
	= 1.501	= 1.115	`= 1.721
Swept Volume (V) cc	1,997	1,999	1,996
Compression Ratio (R)	11	11	11?
Peak Power (PP) HP	310	301	350
@ NP RPM	11,000	9,250	12,000
PP/V HP/L	155.2	150.6	175.4
Brake Mean Effective Pressure @ NP (BMPP) Bar	12.63	14.56	13.08
@ Mean Piston Speed (MPSP) m/s	21.01	24.67	20.92
Mean Gas Velocity at Inlet @PP (MGVP) m/s	64.53	76.57	69.16
	4 steel or Ni-alloy valves per cylinder with steel Coil-spring Valve Return System (CVRS)		
Mean Valve Speed @ PP (MVSP) m/s	4.54	3.89	not available (na)
BNP m/s	15.77	13.75	18.00
BMPA/MDR* Adj. Bar	12.90	14.87	13.35

*For petrol engines, where:-

Air Standard Efficiency = ASE = $[1 - 1/(R)^{0.4}]$, then

$$\frac{BMPA}{MDR} = \frac{BMPP}{MDR} \times \left(\frac{ASE\ @\ R = 12}{ASE\ @\ \text{specified}\ R}\right) = 24 \times (EV \times EC \times EM) \quad \text{Adjusted Bar;}$$

where:- EV = Volumetric efficiency;
EC = Combustion efficiency;
EM = Mechanical Efficiency.

The reasoning behind this expression is given more fully in Analysis Part 2 Page 8

Grand Prix 1.5L TC

Year	1977	1982	1983	
Make	Renault	BMW	Honda	
Type	EF1	M12/13	RA163E	
Data Sources	571,909	741	573,933	
Configuration Valves per Cylinder @ Included Angle (VIA)	All as for the F2 engines			
Bore/Stroke (B/S)	86mm/42.8 = 2.009	89.2mm/60 = 1.487	90mm/39 = 2.308	
Swept Volume (V) cc	1,492	1,499	1,489	
Compression Ratio (R)	7	6.7	7?	
Manifold Density Ratio (MDR)	2.5	2.58	na	
Peak Power (PP) HP	510	575	600	
@ NP RPM	10,500	10,500	12,000	
PP/V HP/L	341.8	383.6	403.0	
Brake Mean Effective Pressure @ NP (BMPP) Bar	29.13	32.69	30.05	
@ Mean Piston Speed (MPSP) m/s	14.98	21.00	15.60	
(MPSP relative to value of F2 NA)	(-29%)	(-14.9%)	(-25.4%)	
Mean Gas Velocity at Inlet @PP (MGVP) m/s Note (1)	46.01	65.18	51.58	
	4 steel or Ni-alloy valves per cylinder with steel Coil-spring Valve Return System (CVRS)			
Mean Valve Speed @ PP (MVSP) m/s	na	na	na	
BNP m/s	15.05	15.61	18.00	
(BNP relative to value of F2 NA)	(-4.6%)	(+13.5%)	(0%)	Note (2)
BMPA/MDR Adj. Bar	13.57	14.98	na	
(BMPA/MDR relative to value of F2 NA)	(105.2%)	(100.7%)	na	Note (3)

Note (1): assuming same Inlet Valve Head Diameters (IVD) in the Renault and Honda TC engines as in the NA.

Note (2): Speed-Limiting factor

It will be seen that a limiting Mean Valve Speed in the TC engines, represented by the surrogate parameter 'BNP' (see Note 13 Part III), was controlling the value of NP in the cases of the highly over-square Renault and Honda TC engines to the same level as the NA engines. Therefore, TC powers were not as high as might have been expected.

Note (3): BMPA/MDR: TC versus NA

The similarity of this value for the BMW suggests that the TurboCharger was not very efficient, since the TC value should have been relatively higher, as is the Renault TC (see Note 96).

Constructional features of NA and TC engines

CH1 and EF1 had belt-driven camshafts.

Renault and Honda had wet Al-alloy Nikasil-coated liners in thinwall cast-iron blocks.

BMW had cast-iron unlinered blocks taken from high-mileage production 89mm bore engines from which all the residual casting strains had relaxed. Therefore, after boring to 89.2mm the bores stayed perfectly round and so minimised friction.

Note 95
Best 3.5L NA engines in 1987 and 1988

Year	1987	1988	
Make	Cosworth	Cosworth	
Type	DFZ	DFR	Note (1)
Sources	47,62,207	47,62,207	
Configuration	90V8	90V8	Note (2)
Valves per Cylinder	4v/c	4v/c	Note (3)
Bore/Stroke (B/S)	90mm/68.6 = 1.312	90mm/68.6 = 1.312	
Sweptvolume (V) cc	3,491	3,491	
Compression Ratio (R)	12	n.a.	
Peak Power (PP) HP	565	594	+5.1%
@ RPM (NP)	10,500	10,750	
Brake Mean Effective Pressure @ NP (BMPP) Bar	13.8	14.2	
@ Mean Piston Speed (MPSP) m/s	24.0	24.6	
Mean Gas Velocity at Inlet @ PP (MGVP) m/s	76.0	72.3	
Mean Valve Speed @ PP (MVSP) m/s	na	na	
BNP m/s	15.8	16.1	
Weight (W) kg	145	140	
PP/W HP/kg	3.90	4.24	+8.9%
Installed in:-	Tyrrell* 016	Benetton B188	
Most successful driver	Jonathan Palmer**	Thierry Boutsen	
Championship points as ratio of Winner:-			
Constructor	8.02%	19.6%	
Driver	9.6%	30.0%	

*Winner of the Colin Chapman Cup.
** Winner of the Jim Clark Cup.

Note (1):- The 'DFZ' clearly was expected to be the last of the line, following DFV (Double Four Valve), DFW, DFX, and DFY, so perhaps 'DFR' stood for 'Re-Born'!

Note (2):- The DFR was 80% different in parts from the DFZ (207).Taking advantage of the new 5½ inch diameter carbon-carbon clutch instead of the previous 7¼ inch, the crankcase was redesigned to lower the crank by 1 inch (574).

Note (3):- Cosworth had an agreement with Yamaha by which the Japanese firm would design 5v/c cylinder heads (3 inlet, 2 exhaust) along the lines which had been quite successful since 1985 in racing-sports motor-cycles (FZ750) and which were used for 1986 in Japanese F2 (OX66 V6 2L). A forecast of 630HP had been given for this joint-project DFR.

No such power was obtained in winter 1987-1988 tests. Therefore the heads had to be re-designed rapidly to normal Cosworth 4v/c principles (207).

It is speculated that Honda, when deciding to develop their 1.5L TC engine for ` 1988, anticipated that the 3.5L NA Cosworth-Yamaha would *not* produce the forecast power, since their V4 750cc 4v/c racing-sports motor-cycles usually beat the IL4 750cc 5v/c Yamahas. The ageing Honda F2 engine also beat the new Yamaha F2 in the final Japanese series of 1986.

Note 96

Effects of Pressure-Charging on the Power equation

When a piston engine is Pressure-Charged (PC) it affects the factors of the basic Power equation, as set out in Note 10, in a variety of ways (in the following '1' refers to the state at the air intake, '2' to cylinder inlet and '3' to cylinder exhaust. Temperatures (T) are absolute, i.e. degrees Kelvin so that Standard ambient temperature of 15 Celsius is 288 Kelvin. Pressures (P) are also absolute, i.e. P1 = Standard ambient pressure = 1 Atmosphere Absolute (ATA) = 14.7psi and 'Pressure at the Inlet Valve' (IVP) = P2 is in ATA) :-

Manifold Density Ratio (MDR) is increased by: IVP x (T1/T2);
and T2 depends upon:- IVP; the efficiency of compression;
and the cooling before cylinder inlet (2) from fuel evaporation and any interposed heat-exchanger (intercooler).

Volumetric Efficiency (EV) is raised in 2 ways:-

- By reducing the heat convected into the charge post-cylinder-entry, mainly from the hot inlet valve, because T2 is higher than T1; empirically the effect is close to

  $$\left(\frac{EV(PC)}{EV(NA)}\right) = \sqrt{(T2/T1)}$$ Ref. (594,Fig.141).

- By reducing the volume of residual exhaust gas in the combustion chamber because P2 is higher than P3; the effect is

  $$\left(\frac{EV(PC)}{EV(NA)}\right) = \left(\frac{R - (P3/P2)}{R - 1}\right)$$ Ref. (121).

  where R = Compression Ratio.

  When the engine is Mechanically-Supercharged (MSC) P3 can be taken as P1, so that

  $$\left(\frac{EV(MSC)}{EV(NA)}\right) = \left(\frac{R - 1/IVP}{R - 1}\right);$$

  If the engine is Turbo-Charged (TC), P3 is greater than P1 by the back-pressure needed to drive the turbine and the full expression can then be reformed as

  $$\left(\frac{EV(TC)}{EV(NA)}\right) = \left(\frac{R - RT/IVP}{R - 1}\right)$$

  where RT = expansion ratio across the turbine.

Mechanical Efficiency (EM) may be reduced or increased:-

- With MSC EM is **reduced** by the power extracted from the crank to drive the compressor ***less*** the pneumatic recovery of power in the cylinder on the inlet stroke because P2 is greater than P1. This recovery is much less than the supercharger driving power because of the inefficiency of the compressor and other factors described below, hence the net reduced EM.

  The in-cylinder recovery is in itself a very complex situation since:-

  (i) there is a drop of total pressure (static pressure plus kinetic energy)across the inlet valve;

  (ii) with the inlet valve opening *before* Top Dead Centre (a timing universal after 1914, to take advantage of exhaust exit momentum) and always closing after Bottom Dead Centre (to take advantage of inlet charge momentum) the P2 effect is initially and finally *resisting* a rising piston;

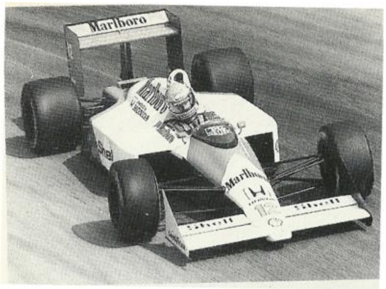
  (iii) if there are tuned individual inlet tracts – not known with MSC in this review - there will be resonance effects to enhance P2.

  (Of course, effects (i) and (iii) are common to NA as well as MSC).

- With Turbo-Charging (TC) EM is **increased** for the following reason:-
  the back pressure in the cylinder has to rise to P3 so that the pressure drop (P3 – P1) can drive the turbine which in turn drives the compressor to produce the inlet pressure rise (P2 – P1). This increase in P3 subtracts power pneumatically from the crank on the exhaust stroke with a rising piston (subject to similar complex effects as listed above for inlet power recovery). However, because the exhaust temperature is around 1,000C thermodynamics means that (P3 – P1) is less than (P2 – P1) (depending on overall machine efficiency). Therefore P2 is greater than P3 and puts more power pneumatically into the crank on the inlet stroke than is deducted by P3 on the exhaust stroke. As there is no mechanical power subtraction for TC the value of EM is increased.

Note 97

McLaren-Honda problems at Monza in 1988



The Italian GP at Monza in 1988 provides a very interesting example of a racing team's engine-related problems and their management.

The Honda RA168E 1.5L TC powered the 2 McLaren MP4/4s of Alain Prost and Ayrton Senna and also the 2 Lotus 100Ts of Nelson Piquet and Satoru Nakajima. Senna and Prost were 1st and 2nd on the grid, Piquet 7th and Nakajima 12th.

The McLaren drivers led the 51 lap race at first in grid order but Piquet spun off at 11 laps and Nakajima had engine failure at 14 laps.

Prost's engine had an audible misfire from lap 29 onwards (941), afterwards reported as a faulty sparking plug, which was apparent in the pits from telemetry (574). Prost drove on, however, still 2nd until a piston collapsed on lap 35 (574), when he then pitted.

After 2 engine failures the Honda technicians then radioed Senna, who had about half-a-minute lead over Gerhard Berger (Ferrari 187/88C) who had inherited 2nd place, and advised him to richen his mixture by the in-cockpit adjustment (941). With his fuel status displayed on-board this obliged him to slow down to make his 150L of fuel last the 305 km of the race.

Observing Senna to be going more slowly, the Ferrari pit radioed Berger to speed up and on starting lap 49 the gap was reduced to a few seconds and Senna could see the Ferrari in his mirrors.

Coming upon the inexperienced Jean-Louis Schlesser (WilliamsFW12-Judd, NA 3.5L), a lap behind, at the 1st chicane after the start line, Senna felt obliged to try to pass and trust Schlesser to make room – *which he did not do!* The cars collided. Schlesser apologised afterwards but the McLaren's race was finished with 2 laps to run, during which Senna might or might not have been able to stave off Berger. The Ferrari driver won and his team-mate was 2nd, to the especial delight of the *tifosi* because it was the first Ferrari win since the recent death of Enzo.

Note 98

Early V10 racing-engine projects

Porsche

A “rally” from Berlin to Rome was scheduled for September 1939 to celebrate the “Axis” of collaboration between their countries and Porsche were contracted to build special cars for this event (which was really a road-race like the Mille Miglia). These cars were based on Volkswagen running gear but having a tubular chassis with streamlined 2/3 seater bodies. Prototypes were fitted with modified higher-power versions of the 1L engine provided for the VW.

Meanwhile a thoroughbred racing engine was designed for a mid-engine mounting, type 114, of 72V10 configuration with B/S = 58/56.5 = 1.027, V = 1,493 cc. It had 2 v/c at VIA = 86^0, IVD/B = 0.61, hairpin valve springs, DOHC, a roller-bearing bottom-end (Hirth built-up crank) and NA by triple carburettors (320, 975). See Fig. N98A on P.2.

It is believed that this engine was not in fact built but, in any case, the invasion of Poland by Germany on 1 September 1939 prevented the race from being run.

This Porsche design was the 1st automobile full-racing V10 configuration.

It is interesting that, when Porsche were contracted by Cisitalia just after WW2 to build a Grand Prix 1.5 L PC engine, type 360, they did not pick up this 72V10 1.5L NA design but adopted an F12 layout.

BMW

Paul Rosche has reported that, at the end of 1975 when BMW considered entering the Grand Prix arena he received the order to design a suitable engine.

Considering the Cosworth DFV V8 and the Ferrari 312B F12 competing at that time he envisaged a 3L V10 as a suitable compromise. It was planned to use this engine also in a racing “supercar”. However, the GP project did not proceed then and the V10 was too costly for the alternative installation, which became the M1 Powered by an IL6 3.5L unit (971).

When BMW did decide to enter Grand Prix racing it was with a TC version of their Formula 2 engine, as described in its place in this review. The works engines were withdrawn at the end of 1986. After another decade (September 1997) Rosche was once again given the go-ahead for a 3L NA GP engine. Ref. (1095) describes how 8, 10 and 12 cylinder configurations had already been evaluated and V12 engines built for research but the final choice was V10, built for the 2000 season. Perhaps BMW had divined that the FIA would rule that only V10 engines would be allowed in 2001 and onward (until 2006).

Alfa Romeo

The origin of this 3.5L V10 engine has been mentioned in the 3rd NA section. Fig.N98B on P.2 provides an illustration.

Fig.N98A
1939 Porsche type 114
72V10 58/56.5 = 1.027 1,493 cc
Two-stage shaft drive to each camshaft pair.
DASO 320

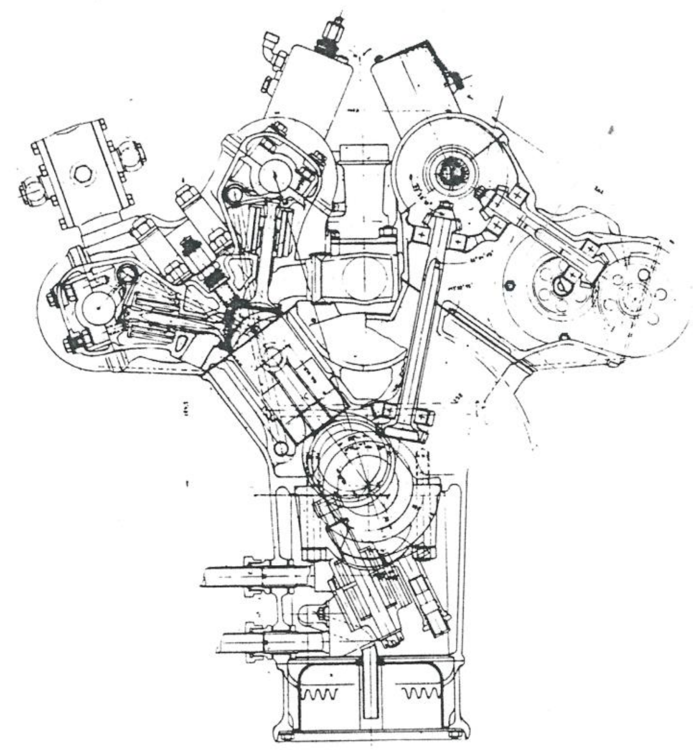
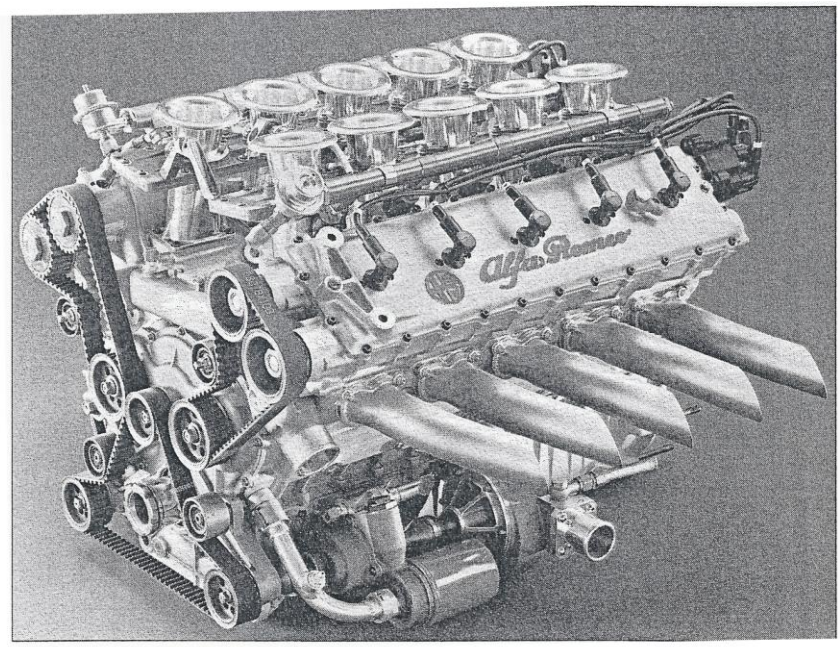


Fig. N98B
1985 Alfa Romeo type 1035
72V10 88/57.52 = 1.53 3,498 cc
Belt drive to camshafts.
Both 4 v/c and 5 v/c heads were examined.
(DASO 1111: Alfa official data via courtesy of John Cundy 30 May 2012).
DASO 1112: Wikipedia quotes power in 1986 as:- 620 BHP @ 13,300 RPM.
DASO for figure: Not recorded.



Note 99

Friction-and-Pumping Mean Effective Pressure (FPMEP) for 4-strokes

Brake MEP = Indicated MEP - FPMEP

The author established in a private study (DST 23 April 1996) that a reasonable correlation for FPMEP for 4-stroke piston engines had the form:-

$$\text{FPMEP} = \text{KF1} + \text{KF2} \times N \times (\text{MPS})^2.$$

With FPMEP in Bar; N = crank speed RPM; and MPS = Mean Piston Speed in m/s:-then the coefficients of correlation were:-

for **Touring** engines KF1 = 1; and KF2 = $25/10^7$.

for **Racing** engines KF1 = ¾; and KF2 = $9/10^7$.

An attached chart 47/dst on P.2* illustrates the data examined for this correlation (data collected up to mid 2001 was about 70% in support). The formulae above have been included in Appendix 1 so as to calculate an Estimated Mechanical Efficiency (EEM, at Line 105) for each CoY example. In these calculations for pre-WW1 engines the values of KF1 and KF2 (shown on Lines 102 & 103) have been chosen as though they were **Touring** units.

The reasons for **Touring** engines having much higher KF coefficients than **Racing** engines are:-

- Longer life required: this demands more material to reduce stresses so that there is more inertia in reciprocating parts, greater piston-cylinder reaction and more oil-shearing area;
- Lower noise required: therefore smaller clearances are required between piston and cylinder so again there is greater oil-shearing loss;
- Lower oil consumption required: Requiring more and higher-radial-pressure piston rings so once more higher oil-shearing loss;
- More Accessories required : self explanatory
- Lower Volumetric Efficiency: *in the past* (see below) the inlet and exhaust systems of **Touring** engines have been much more tortuous than **Racing** engines;
- Lower cost required: **Racing** engines have always had much more labour spent on them to machine parts accurately and fit the parts truly.

The last two factors will be less significant now than in the past because **Touring** engines now have at least “Semi-tuned” inlet and exhaust systems and are produced to a higher quality of machining and assembly.

Racing engines have also had special anti-friction features compared to **Touring** engines, Egs. More ball-and-roller bearings throughout; rollers in valve gear; and more recently low-friction coatings on rubbing parts.

Dimensional Correlation

To obtain an FPMEP correlation with correct dimensions the (N) term should be separate from the $(\text{MPS})^2$ term. The former term when multiplied by Oil Viscosity would represent a Pressure. The latter term when multiplied by a Density would also represent a Pressure. The equation would then be balanced dimensionally.

However this desirable separation of terms could not be identified with the data available.

The accuracy of the correlation therefore must be in doubt beyond $(N) \times (\text{MPS})^2 = 90 \times 10^5\ (\text{RPM}) \times (\text{m/s})^2$.

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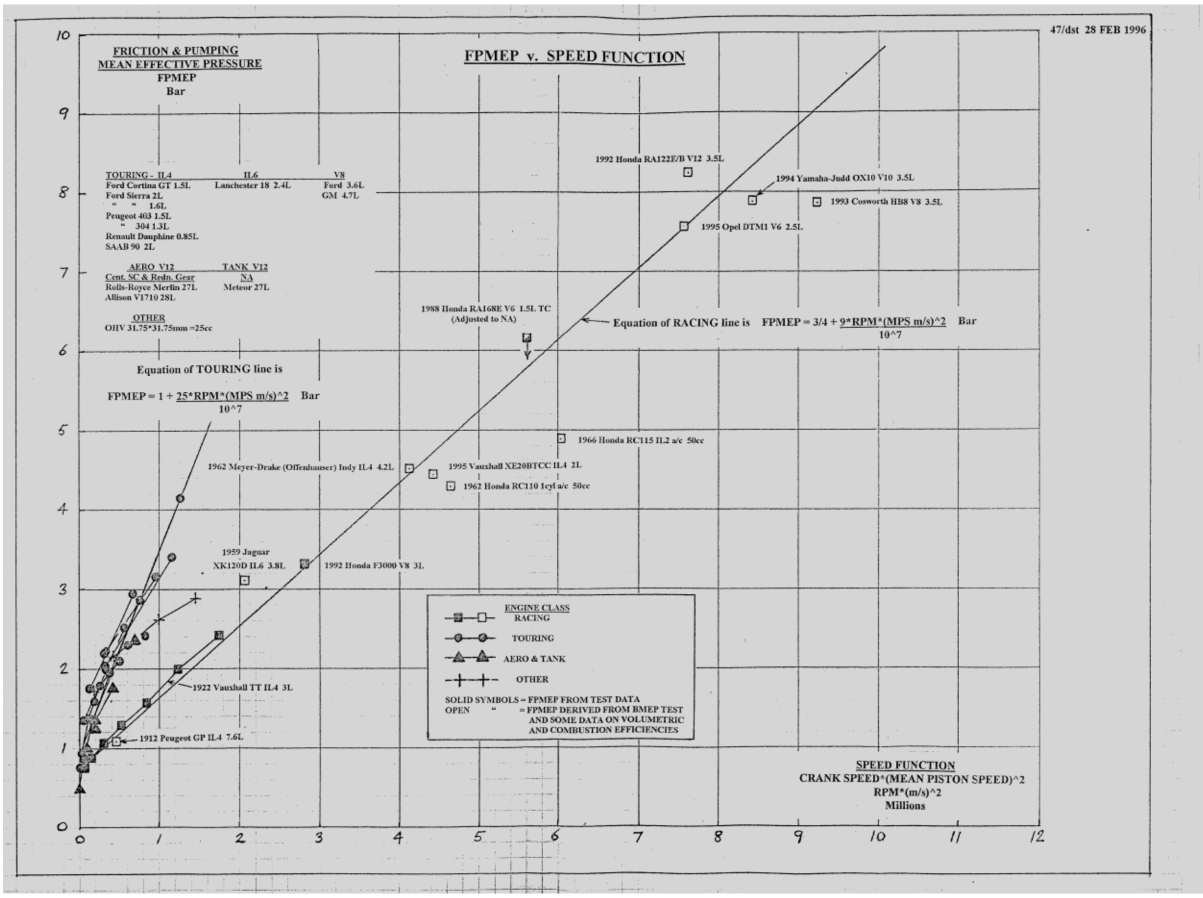
*This chart needs to be enlarged to 200% to be read clearly.


47/dst 28 FEB 1996
FRICTION & PUMPING
MEAN EFFECTIVE PRESSURE
FPMEP
Bar
FPMEP v. SPEED FUNCTION
TOURING - IL4
Ford Cortina GT 1.5L
Ford Sierra 2L
" " 1.6L
Peugeot 403 1.5L
" 304 1.3L
Renault Dauphine 0.85L
SAAB 90 2L
IL6
Lanchester 18 2.4L
V8
Ford 3.6L
GM 4.7L
AERO V12
Cent. SC & Redn. Gear
Rolls-Royce Merlin 27L
Allison V1710 28L
TANK V12
NA
Meteor 27L
OTHER
OHV 31.75*31.75mm =25cc
Equation of TOURING line is
FPMEP = 1 + 25*RPM*(MPS m/s)^2 / 10^7 Bar
1992 Honda RA122E/B V12 3.5L
1994 Yamaha-Judd OX10 V10 3.5L
1993 Cosworth HB8 V8 3.5L
1995 Opel DTM1 V6 2.5L
1988 Honda RA168E V6 1.5L TC
(Adjusted to NA)
Equation of RACING line is FPMEP = 3/4 + 9*RPM*(MPS m/s)^2 / 10^7 Bar
1966 Honda RC115 IL2 a/c 50cc
1962 Meyer-Drake (Offenhauser) Indy IL4 4.2L
1995 Vauxhall XE20BTCC IL4 2L
1962 Honda RC110 1cyl a/c 50cc
1959 Jaguar
XK120D IL6 3.8L
1992 Honda F3000 V8 3L
1922 Vauxhall TT IL4 3L
1912 Peugeot GP IL4 7.6L
ENGINE CLASS
RACING
TOURING
AERO & TANK
OTHER
SOLID SYMBOLS = FPMEP FROM TEST DATA
OPEN " = FPMEP DERIVED FROM BMEP TEST AND SOME DATA ON VOLUMETRIC AND COMBUSTION EFFICIENCIES
SPEED FUNCTION
CRANK SPEED*(MEAN PISTON SPEED)^2
RPM*(m/s)^2
Millions
0
1
2
3
4
5
6
7
8
9
10
11
12

Note 99B
Friction-and-Pumping Mean Effective Pressure (FPMEP) for 4-Strokes
Additional experimental data

Note 99 gave a simple correlation of 4-Stroke FPMEP in the terms:-

$$FPMEP = K1 + K2.NP.(MPSP)^2$$

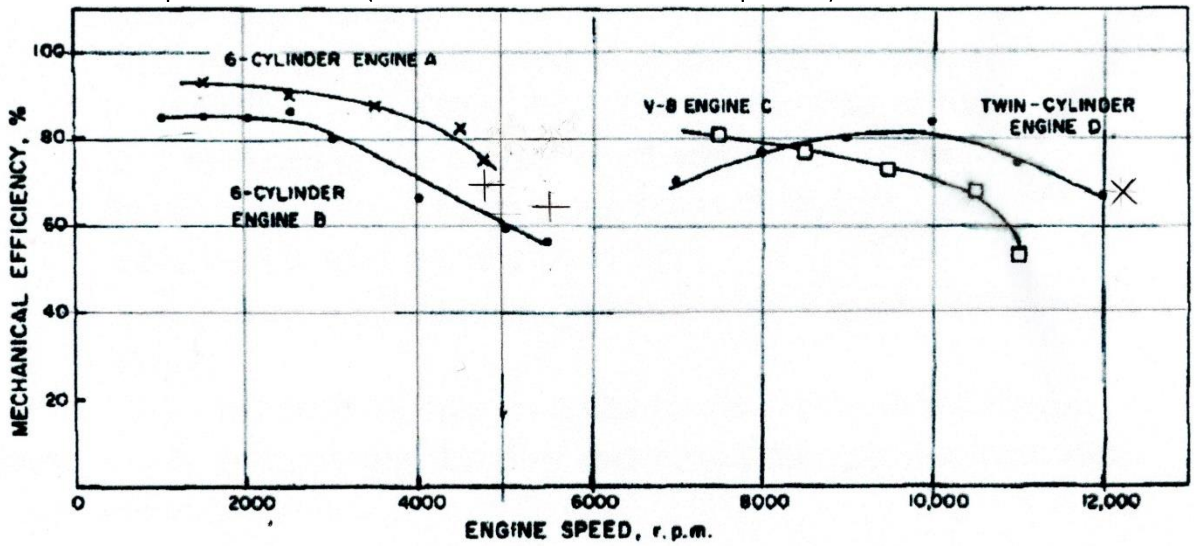
where NP = Peak RPM; MPSP = Mean Piston Speed @ NP (m/s);
for Racing engines K1 = 0.75 Bar; K2 = $9/10^7$;
for Touring engines K1 = 1 Bar; K2 = $25/10^7$;
and FPMEP is in Bar.

The reason for the differences in K1 and K2 between Racing and Touring engines is given in Note 99.

The FPMEP correlation has been used in the standard data and analysis tabulations of the Appendices of this website to provide an Estimated Mechanical efficiency (EEM) at Row 105.

Shell Research data

SAE 68765 in DASO 1230 (see ref . below, advised by courtesy of correspondent Stephen Cansick) provided at its Fig. 2 some Mechanical Efficiencies (EM) v. RPM for 2 Racing and 2 Touring engines. This chart is reproduced below (with additions which will be explained):-



Although the engines tested were identified only by letters A -D and no other data was given, three of the units can be identified and the appropriate data used to calculate EEM for comparison with the experimental figures. This has been done. The 4th ("B") is a matter for speculation. Calculations for EEM are given in Appendix 9.

Comments

Engine A

The 4,800 RPM peak point identifies this engine as:-

1964 Rolls-Royce FB60 (unit fitted in the Austin VanDenPlas Princess R).
IL6 3.75''/3.60'' [95.25 mm/91.44] = 1.042 3,909 cc. 175 BHP @ 4,800 RPM.

Touring K1 & K2 EEM = 70% (shown on the chart as + point) versus test EM = 76%.

A section of the FB60 is shown on P.2.

Engine C

Known to be the 1962 BRM V8 1.5L because Tony Rudd has acknowledged work by Shell Research on this engine.

Clearly, this was an early un-developed unit since EM drops suddenly to 53% @ 10,500 RPM. Its success in powering the 1962 Championships can only have come after that was put right. It seems from the Discussion of SAE680765 that this build, although a dry sump engine, had the crankshaft dipping into a pool of un-scavenged aerated oil in the crankcase.

As finally developed for 1962 the engine gave 197 BHP @11,000 RPM, for which EEM = 72% is calculated (Appendix 1).

Engine D

From information given to the author by the late Brian Lovell, former Managing Director of Weslake Research, it is known that this engine was that firm's WR22.

1964 Weslake WR22 Shell-financed, based on the 1962 BRM cylinder dimensions in order to have a direct comparison with that maker's engine, BRM (Owen) having bought a part share in Weslake. It was the first engine of the "Four-valve Renaissance", with a pioneering 32^0 Valve Included Angle (VIA) between 4 Valves-per Cylinder (4v/c). See "How many valves per cylinder" at P.10.

IL2 68.5 mm/50.8 = 1.348 374.4 cc. 59 BHP @ 12,200 RPM.

Racing K1 & K2 EEM = 68% (shown on the chart as X point) versus test EM = 67% @ 12,000 RPM.

Engine B

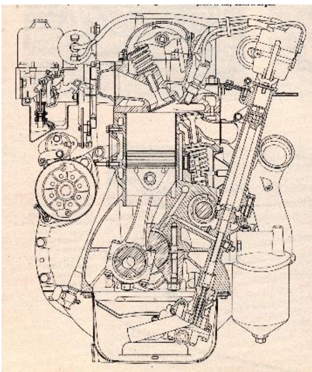
It is speculated that a 6 cylinder engine revving to 5,500 RPM pre-1968 could have been the Bristol 100C, the unit usually fitted in the type 450 production car. Details for this have been calculated.

1953 Bristol 100C

IL6 66 mm/96 =0.688 1,971 cc. 118 BHP @ 5,500 RPM.

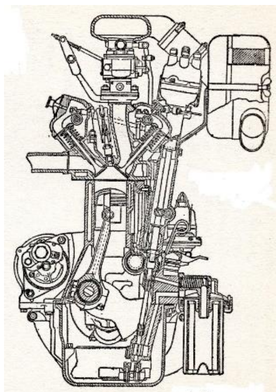
Touring K1 & K2 EEM = 65% (shown on the chart as + point). The test EM for Engine B was 57%.

A section of the 100C is shown below.



Rolls-Royce FB60

DASO 865



Bristol 100C

DASO 337

Conclusions

Note 99, used in the website data tabulations to calculate EEM, fits the Racing engine WR22 very well. For the Touring-class engine FB60 the value of EEM is "in the right ball-park", tho' 6%points low. If Engine B *was* the Touring-class Bristol 100C, then being too high by 8%points may be partially a result of the 100C EM being lowered by the extra rockers in its BMW 328-type valve gear.

Reference

DASO 1230 Some Applications of Basic Combustion Research to Gasoline Engine Development Problems G. Harrow..Shell Thornton Research Centre 1968.

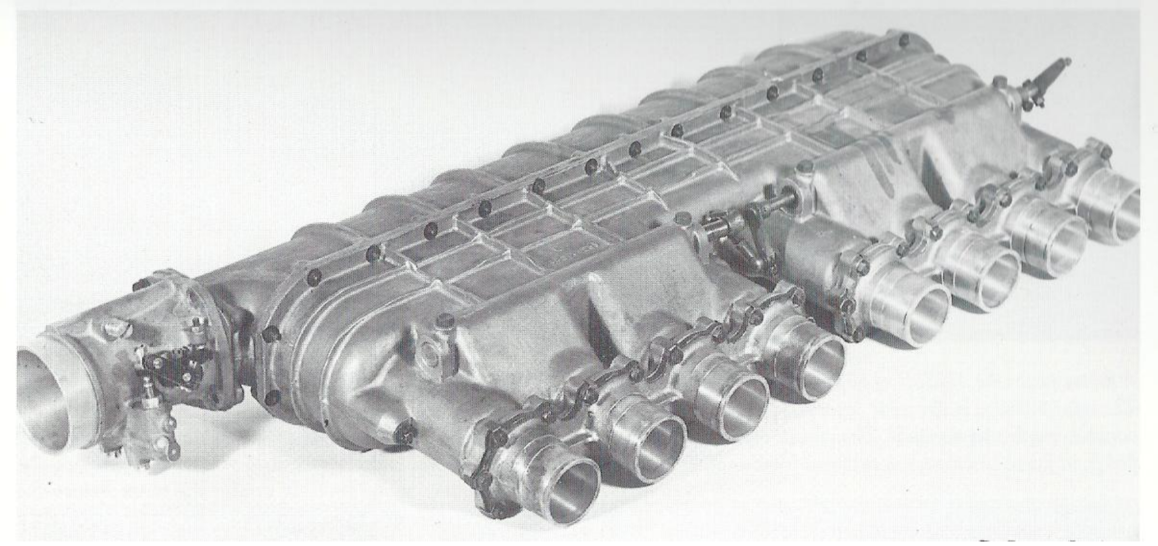
Note 100

Mercedes-Benz experiments with variable-length inlet tracts

Ref. (468) describes how Mercedes-Benz tested on the bench an experimental variable-length inlet tract design for the proposed 1956 300SLR Racing-Sports engine. This was to obtain better torque at below-Power-Peak RPM by retuning the resonant frequency, at some cost in weight and complexity increases.

While this was successful when operated manually, it still required an automatic control system when M-B participation in racing was halted.

Fig. N100A
1955 Mercedes-Benz M196.I (300SLR)
Variable-length inlet tracts.
The length variation was inside the plenum chamber.
DASO 468



Note 101

Traction and Launch Control

“Traction Control” was 1st fitted to a winning Grand Prix car by Williams in their 1992 FW14B-Renault RS3C.

It worked by comparing the measured RPMs of a front wheel with a rear so that any spin was detected and then the Engine Control Unit (ECU) cut the ignition to 1 or more cylinders to control that spin to the %age which gave maximum traction.

This %age depended on the type of tyre. In the mid ‘80s a figure of 15% was quoted by Niki Lauda (571) for an optimum standing start, driver-controlled at that date, of course.

In 2004 5% was quoted (1027) for tyres which undoubtedly had a much higher Coefficient of Friction (approaching 2 (987)) than 20 years previously.

The difference in opinion between McLaren and Cosworth in 1993 over the best way to reduce torque for Traction Control (ignition cut-out versus throttle reduction) is described in Eg. 77.

When Traction Control was later combined with a fully-automatic gear-change (FAGC) then “Launch Control” for optimum Grand Prix standing starts was produced.

However, both Traction Control and FAGC were banned from the start of 1994 until the end of the period covered by this review.

Note 102

The 1993 Technical Rule changes

The Williams team failed to send their 1993 entry paperwork to FISA until 2 days after the November 1992 closure date for the next season. Despite some other team's objections the President of FISA (Max Mosley*) accepted the entry nevertheless on his own authority in February 1993. He took this unilateral decision in the interests of good competition and made this the same reason at the same time for deciding on a group of major technical changes to take effect in 1994.

This was actually a breach of the then-current "Concorde Agreement" between the FIA and the Formula One Constructors' Association (FOCA) which laid down that such changes had to be agreed with all signatories and promulgated with not less than a full season's lead time, *unless* covered by the over-riding clause which allowed unilateral shorter-lead-time changes for *safety* reasons.

The Mosley proposals, banning active (or "reactive") suspension, traction control, automatic gear-changing and anti-lock brake systems (ABS), did *not* have this cover.

The FISA President was reported as admitting that this *could* be seen as a *legal* breach but also saying that he did not regard it as a "*moral*" breach (998).

He is said to have accepted that road cars should benefit in safety from the "driver aids" to be banned from racing cars but to have argued that racing was suffering from them. *If* correctly reported this could only lead to the deduction that racing cars had been made too safe so that they no longer skidded round the track and thereby the spectacle for the paying public of Formula 1 was reduced. It *could* be concluded further that direct profit had become more important to FISA than technical progress – a turning point for Grand Prix racing.

FISA action against active suspension in 1993

The Williams and McLaren teams, having made a success of their large investments in the things to be banned, had most to lose. This was deliberate, on the theory that their less-advanced rivals could catch up and also improve the paying spectacle.

When these teams did not accept these unilateral proposals for 1994 immediately, the FISA Technical delegate in mid-1993-season then ruled that active suspension and traction control were illegal anyway under the existing rules – something which had escaped his notice in the preceding 22 races!

Williams and McLaren protested against this decision and were over-ruled.

Under this threat of immediate banishment of several teams, they all agreed the February rules in August 1993 – 8 months before the 1st 1994 race.

In-race refuelling re-introduced for 1994

Also at that date in-race refuelling was re-introduced for 1994. It has to be presumed that this was also to increase the spectacle, adding more pit-stops and the possibility of place changes, since overtaking on the road with aero-downforce cars was so difficult.

This re-introduction was after a 10 year ban (1984 – 1993 inclusive) on *safety* grounds.

Technical advances using aircraft refuelling equipment to provide fuel loading without spillage and possibility of fire (theoretically**) were in *this* case acceptable to FISA.

Fund diversion

It can be argued that the restrictions on chassis technology simply diverted more money to engine improvements.

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*Max Mosley was elected as President of FISA in October 1991, displacing Jean-Marie Balestre. He became President of the parent body, FIA, when Balestre stood down from that post in October 1993. Mosley then abolished FISA and reorganised the sporting regulation side to come directly under the FIA via a new World Motorsports Council (WMSC).

**But not in practice. At the 9th race in 1994 (Hockenheim) there was a fuel leakage and fire at a Benetton pit-stop in which the driver (Verstappen) and several mechanics were burned, fortunately not badly.

Note 103

“Diamond-Like-Carbon” (DLC) surface treatment

“Diamond-Like-Carbon” (DLC) was developed first as a highly-wear-resistant surface coating for steel in the ‘80s for tool applications and machinery guides and seals. As its name implies it is extremely hard:- Vickers test figures of 3,000 to 4,000 are quoted (989), compared to 850 for nitride steel and 10,000 for real diamond. This reduces greatly the surface friction in rubbing mechanisms compared to untreated parts.

However, initially, DLC was found to be too brittle and having insufficient adhesion where impact occurred such as in 4-stroke piston engine valve gear. Applying himself to this problem in 1993 Jacques Buchaux (founder in 1969 of the French engineering firm JPX which later turned to producing precision motor-sport components) in collaboration with Claude Lory (founder in 1989 of Sorevi, a French firm already specialising in DLC using their patented “Cavidur” plasma-cum-physical vapour deposition process developed in 1985 by Limoges University) developed a multi-layer technique which overcame the brittleness and lack of adhesion of the original coating. This began at the parent metal with a soft, ductile, material and led through stages increasing in hardness to give the required working surface, the combined layers still aggregating only around 4 microns thickness.

A first successful application of the new process was on the inverted-cup tappets of the 1993 Peugeot DOHC 80^0V10 3.5 L Le Mans engine (978), replacing a previous hard chrome coating (989).

JPX had been supplying PVRS sleeves to Renault for the RS series since 1991 (989) and therefore it seems extremely likely that they applied DLC to such parts from 1994.

By 2004 it was possible to provide DLC coatings (giving a shiny black surface finish) to steel alloys on parts such as :-

Camshafts; tappets; finger followers; PVRS cylinder bores and their pistons;
gudgeon pins (allowing deletion of bronze bushings); and gears.
Also to Ti-alloys (valve stems).
Also to Al- and Mg-alloys.

It was stated by Buchaux that it would have been impossible to run 2004 Grand Prix valve gear speeds (MVSP around 10 m/s) for more than a few minutes (even in a well-lubricated environment) without DLC* but with it there is no limit. Apart from the life extension there is a reduction in the friction power losses. Ref. (1024) states that in a MotoGP motor-cycle racing engine a gain of 8HP was made by coating the cams and followers with DLC – this would represent +3% or more of power output.

Development work was in hand in 2004 to apply DLC to piston rubbing surfaces (978).

The sources do not indicate what DLC coatings cost.

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*In 2000 at the Indianapolis GP circuit, which permitted a wide-open throttle for 22 seconds along the pits straight, a longer time than any other venue, Hakkinen’s Ilmor-Mercedes FO110J engine blew up on this section at ⅓rd distance after an acknowledged finger-follower coating (almost certainly DLC) adhesion failure (700).

Note 104

Relation of Season-average Lap Speed to Power and Weight, 1993 – 1994

Power relation

During 1993 informed opinion* was that the Renault RS3/4 engines in the Williams FW15C had an average of 50 HP more than the Ford-Cosworth HB5/6 units in the main rivals, the McLaren MP4/8 and the Benetton B193. All these cars had active suspension and various electronic "driver aids".

This provided an opportunity to compare Lap Speed differences with Power difference.

Over the 1993 season the Lap Speed differences at each race in Qualification were as follows, the Williams best speed being compared with the best Ford-powered figure:-

*Of the late Brian Lovell, former Managing Director of Weslake Developments.

Lap Speed %age lower than Williams

Race No.	1	2	3	4	5	6	7	8
McLaren	[0.1]	2.3			2.4			2.4
Benetton	2.1		2.2	2.2		0.8	2.3	

Race No.	9	10	11	12	13	14	15	16
McLaren					1.8	1.4	[0.1]	[-0.6]
Benetton	1.8	0.8	0.8	1.4				

The [bracketed] entries were discarded: Race 1 at Kyalami by Senna being "too good to be true" and the Benetton figure used; Races 15 and 16 because by then Prost driving the Williams had clinched his 4th Championship at the 14th race and announced his retirement would come at the end of the season. He simply did not need to try as hard as before. The No.2 Williams driver, Damon Hill, had not raced at either of the last 2 circuits (Suzuka and Adelaide) and spun in Qualification at both of them.

The season-long average of the 14 results judged not to be affected by the special factors mentioned was therefore:-

1.8% advantage to Renault over Ford-Cosworth.

A difference of 50 HP was 50 HP/700 HP = 7.1%.

The relation is therefore:-

7.1/1.8 = + 3.9% Power provides + 1% of Lap Speed.

Weight relation

At the 1st race of 1994 at the Interlagos circuit of Brazil, it was reported in the TV commentary that the difference between carrying 200 litres and 70 litres of fuel was worth 3 seconds per lap in 76 seconds (Pole time), i.e. -4% of speed.

The 130 litres of petrol, weighing 94 kg in a car whose weight would be then:-
505 kg empty of fuel + 70 kg for the driver + ½ of the 70 kg during Qualification running = 610 kg, represented 15% weight difference.

The relation is therefore:-

15/-4 = + 3.8% Weight produces -1% Lap Speed.

Overall

A simple rule of thumb for average effects over the 1993 – 1994 season was to take

+4% of Power or Weight gives a 1% effect on Lap speed,
Plus for Power; *Minus* for Weight.

1997 – 2008 Results

Analysing a sample of 13 computerised predictions of Weight/Lap Speed sensitivity supplied to TV commentators by the teams over the years 1997 to 2008 showed a similar relation to the above for Weight, i.e. +4% weight loses 1% of Lap Speed.

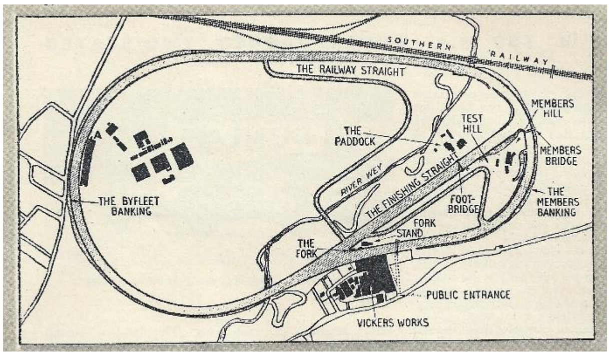
Note 104B
Brooklands Outer Circuit Lap Speeds

While seeking data on the power of the famous Freddy Dixon modified NA 2 Litre Riley, which he never bench-tested, a look was taken of its speed on the Brooklands Outer Circuit in comparison with other cars. This led to a correlation of speeds and powers on that track and it is thought that this might be of interest to others.

The circuit

Brooklands was built in 1907, the first dedicated motor-racing track in the world. Fig 1 shows its plan.

Fig. 1



DASO 331

The Outer Circuit, lapped anti-clockwise, comprised a reinforced-concrete uniform 100 feet wide track in the following sections:-

- The Home (or "Members") Banking with an inner radius of 1,000 feet turning through 165^0, rising from level to banking whose cross-section increased smoothly in a curve to a top quarter at 30^0 to the horizontal, the banking then decreasing to level at the Railway Straight;
- A 9^0 curve entering the Railway Straight;
- The level Railway Straight;
- The Byfleet Banking with an inner radius of 1,500 feet turning through 213^0 and a top angle of 20^0, with the same type of transition sections at entrance and exit;
- A level section with the Fork curve 27^0 to the right, where a Finishing Straight continued straight on (feature never copied on any other motor-racing track)..

The total turning of 360^0 was therefore made up of:_ 165^0 LH + 9^0 LH + 213^0 LH – 27^0 RH
The track was 2 miles 1,350 yards long on the centreline (2.767 miles; 4.453 km).

It was reported to have been designed for 120 MPH (article in *M. Sport* March 1925 by A.W.Phillips). This is discussed in an Appendix, regarding the assumed Coefficient of Friction on the banking.

Lap Speeds

Lap Speeds(LS) from 1910 to 1938 are given in Appendix 10. These are all open-wheeled cars, mostly nominal 2-seater* except where shown as narrow 1-seaters. The Peak Powers (PP) and Laden Weights (W) are given (W is in Hundredweights (cwt), as originally measured; 1 cwt = 112 lb = 50.8 kg). The Laden weight is the quoted empty weight plus a nominal amount to include fluids and driver.

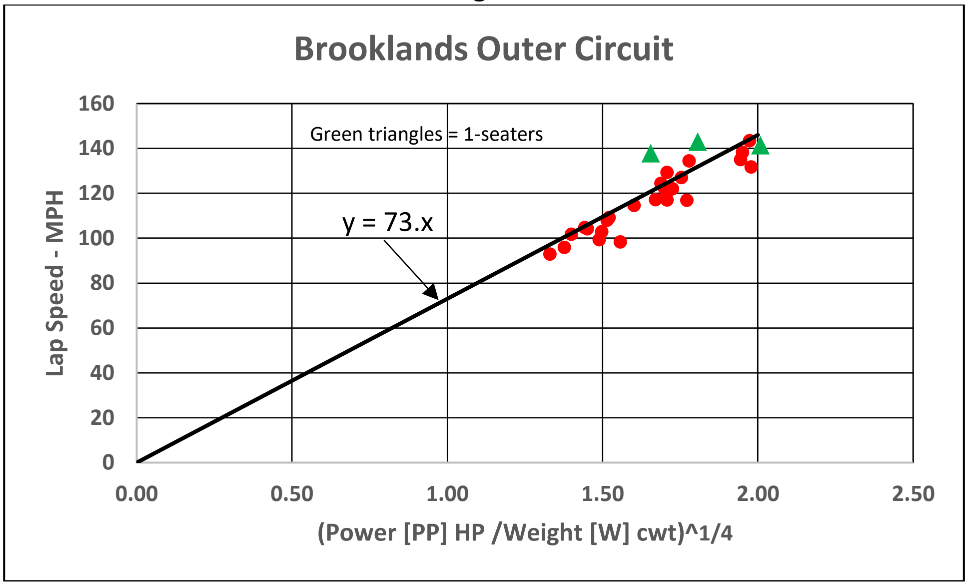
*The Napier-Railton all-time record holder is classed as a "Nominal 2-seater" because of its body width.

Continued on P.2

Relation between LS and PP/W

Power/Weight (PP/W) ratio is a meaningful parameter in relation to Lap Speed (LS) because it has the physical dimensions of ***Speed x Acceleration***. Fig. 2 shows **LS** related to **(PP/W)^$^{1/4}$**.

Fig. 2



Inevitably there is a scatter of data because of:- less-than-scientific accuracy of figures available over 27 years; weather variations, particularly of wind speed, direction and turbulence, especially at the Fork curve after the erection there of large sheds close to the track; and, of course, the driver skill, experience and the determination to cope with the notorious bumps which quite quickly appeared due to settlement.

concerning the latter, it is relevant to quote something written by Tim Birkin, shortly after setting a record of nearly 138 MPH in 1932:-

"I think that it is, without exception, the most out-of-date, inadequate and dangerous track in the world. Brooklands was built for speeds no greater than 120 MPH and for anyone to go over 130 without knowing the track better than his own self is to court disaster. The surface is abominable. There are bumps which jolt the driver up and down in his seat and make the car leave the road and travel through the air." (Quoted in Bonham's literature for the auction of Birkin's record-breaking Bentley UU5871 in 2012.)

The most notorious bump was where the track crossed over the River Wey, towards the end of the Home banking. Here the track had settled on either side of the bridge, which had not settled (as described by H.N.Charles in DASO 331). The resultant effect on fast cars is shown in Fig. 3.

Fig. 3

John Cobb driving his Napier-Railton over the R. Wey bump

With all four wheels off the track there must have been previously a correction to straight line travel to prevent a sideways displacement.

pinterest

Given the mentioned effects on LS, a "Maximum" line for nominal 2-seaters has been drawn on the chart, passing through the all-time record of 143.4 MPH set by John Cobb in 1935, at:-

LS = 73 x (PP/W)^$^{1/4}$ (rounded-up)

Clearly, 2 out of the three 1-seater cars beat the 2-seater "Maximum" by a substantial amount, due to lower drag. This and other good and bad figures are discussed on P.3.

Variations from "Maximum" line

1926 2 Litre 1922 Sunbeam

Below the line by 13%. With due respect to the driver, J. Spencer, he was a private owner and did not figure much in Brookland's annals with a 4 year old car.

1927 Mercedes 1924 GP

About 10% low. One of Ferdinand Porsche's "Hard to drive" designs, with a low polar moment of inertia and very stiff suspension. Two out of 3 built crashed. Mays recorded the comments of Segrave on his run at Brooklands:-"*Ray, you're damn lucky to be alive, and if you take my advice you will never drive that car again!*" (DASO 446). Enough said.

1930 Delage 1927 GP

Very close to the line, but noted because it was *assumed* that this run was at the much-publicised 170 HP. Note 5 discusses in detail the reasons why this could not have been the *sustainable GP-race-long* output because it would require RPM in the piston-ring flutter region. However, the rings *could* have lasted for a few laps of Brooklands.

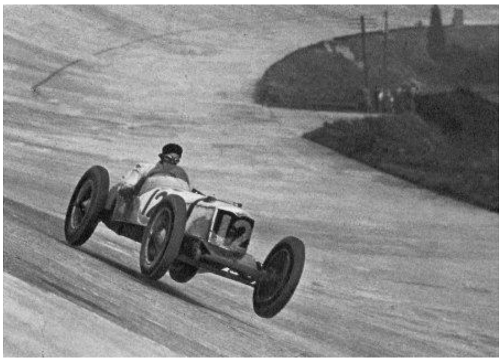
1935 MG Q-type

Below the line by 3%. The power output is the unsustainable short-run level quoted by H.N. Charles in DASO 331. He also commented about the lap speed, by a private owner "*The lightness of touch and degree of anticipation necessary to do this would be possessed by very few people. His driving on that occasion was better than the car he drove.*" Charles' own earlier experience with the Q-type prototype at Brooklands led him to design the all-independently-sprung R-type.

1935 Dixon-Riley

This is 3½% better than the line. It was a car with very low lines for its type, as can be seen in Fig.4.

Fig. 4



Freddy Dixon driving his 2 Litre Riley Special over the R. Wey bump.

It was concluded that the 150 HP conservatively estimated by the car's 1995 owner, Mark Gillies, was a reliable figure. (DASO 141). This was a very remarkable power for a Naturally-Aspirated IL6 2 Litre engine in 1935. It is discussed further in "Significant Other" at SO11.

Unique Cars and Parts USA

1937 Talbot t150C

Nearly 4% above the line. This was one of only 2 cars in the data to have independent front suspension (IFS; the Multi-Union was the other). All other cars listed had rigid axles with leaf springs all round. The T150C rear axle was rigid. This IFS *could* have been an advantage over the Brooklands bumpy surface. It would have depended on whether the resultant low-front-to-high-rear roll axis, the relative front/rear roll stiffnesses and the weight distribution, gave a balanced car or not. Over or understeer would be definite handicaps on the banking. Presumably the T150C was sufficiently balanced.

1938 Multi-Union

Although a 1-seater with a streamlined body and IFS, the Multi-Union lapped 3½% *below* the line officially (*unofficially* it came within 1/10 second of the record, 143.2 MPH, according to DASO 286). Considering that the other 1-seaters lapped so much faster than the line – the Bentley special driven by Tim Birkin +14% in 1932 and the Oliver Bertram-driven Barnato-Hassan + 8½% in 1938, it is not clear why Chris Staniland did not achieve more with the Multi-Union. The question of balance for a car with IFS but a rigid rear axle has been mentioned under the Talbot T150C above. Maybe the Multi-Union in its final configuration did not have the required stability. Very few people in 1938 knew how to provide a car with steering appropriate to the circuits on which it was raced. Continued on P.4.

1938 Multi-Union, continued.

There is another possible explanation for the relatively disappointing Multi-Union speed as a 1-seater – *insufficient weight*. Brooklands surface was in a very poor condition from quite early in its history, as Tim Birkin described (see above). A heavy car would be able to keep its wheels on the ground better than a lighter one. The 1-seaters which were well above the line were 31/32 cwt; the Multi-Union, derived from an original 750kg (14.8 cwt without wheels and tyres) formula car was only about 20 cwt. The Napier-Railton was 33 cwt. The three other 2-seaters with $(PP/W)^{1/4}$ around 2, the same as the Napier-Railton, which were 3 to 9% below the line were around 20 cwt.

The all-time record

The all-time Brooklands Outer Circuit record was set by John Cobb in his Napier-Railton at 143.44 MPH in October 1935. If, for simplicity, it is assumed that this was also the speed on the Home banking*, Appendix A shows that this was consistent with a Coefficient of Friction of about 0.4. Profile 28 records that the surface was "damp". The calculation for the Byfleet banking is similar. This is twice the track design assumption. The tyres lasted just two laps!

As we now know, although very few people would have known in the mid-'30s, a pneumatic tyre in order to develop lateral force must run distorted at a "slip angle" to the car's axis. The car travels at this angle to the vehicle centre-line, which the driver allows for automatically with the steering (until the angle goes beyond a limit set by the tyre's construction and materials). An observer of Cobb's first stab at the record in Easter 1934, placed where he could see a semi-plan view of the car on the Byfleet banking, wrote in *M. Sport* May 1934:-

"*On the Byfleet the Napier-Railton seemed to be in a steady slide, the tail a little higher on the banking than the front*". The car was not actually sliding but the observer very accurately described the effect of slip angle.

With rigid axles at each end - i.e. a horizontal roll axis - and equal spring bases fore and aft (Profile 28 plan), the car presumably was stable – more by Reid Railton's intuition than by calculation it is suspected (perhaps only Maurice Olley could have done the calculation at that date).

The record lap was described by Cobb as "*Like seeing how far you can lean out of an upper-storey window of a tall building without falling out!*".

*The car would have been faster along the Railway Straight – 160 MPH was mentioned – but slower while negotiating the Fork curve with its wind turbulence.

Appendix
Theory and practice of cars on a banked track

The theory for this is available on the internet (HyperPhysics):-

where the radius of the curve is **r ft;**
gravitational acceleration is **g = 32.17 ft/sec^2;**
the angle of the banking is $\boldsymbol{\theta}$ $\boldsymbol{degrees}$;
the Coefficient of Friction is $\boldsymbol{\mu}$;

$$\text{then} \quad V_{max} = \sqrt{\frac{r.g.(\sin\theta + \mu.\cos\theta)}{\cos\theta - \mu.\sin\theta}}$$

in **Ft/Sec (x 60/88 = MPH)**

For the Brooklands Home banking the maximum safe radius for the car Centre of Gravity is, say, 10 ft in from the lip
at:- r = 1,090 ft; and $\theta = 30\ degrees$:-

then	for $\mu =$	0	0.2	0.4	
	V =	97	120	144	MPH

For the Brooklands Byfleet banking
At r = 1,590 ft; and $\theta = 20\ degrees$: –

then	for $\mu =$	0	0.2	0.4	
	V =	93	120	146	MPH

The Brooklands track designer, to arrive at the reported design speed of 120 MPH, seems to have assumed a Coefficient of Friction of 0.2. This appears as exceedingly pessimistic, but there would have been little or no data on the friction of narrow, high-pressure pneumatic tyres at that date. The greater radius of the Byfleet banking allowed a reduction of bank angle to 20^0 for the same speed (which would have saved a lot of earthwork).

The practice resulted in a final Lap Record of 143.4 MPH by John Cobb in 1935 in his Napier-Railton. This corresponds to a Coefficient of Friction of 0.4 (this is assuming for simplicity that the lap speed corresponded to the banking speed).

Not surprisingly, the fastest cars did not race when the track was wet.

Note 105

Ilmor share ownership, 1984 – 1994

Penske and GM

According to (468) Penske originally took 50% of the Ilmor shares but, when he interested GM in having an Indy engine built for them and badged as Chevrolet, he sold ½ of his shares to them. Ongoing financial aid to design, build, test and supply the engines was then provided by GM.

GM and Mercedes-Benz

According to (1007) GM in 1993 had wished to continue with Ilmor on the type 265D engine for 1994. However, they announced their withdrawal in September 1993. Shortly afterwards the Chevrolet General Manager confirmed that “another party” had outbid them with an offer to Ilmor of 3 x the money per year for 2 years longer which made, he said, a difference of $100M in total.

“Another party” was Mercedes-Benz.

Note 106

Single-cylinder engine testing and its drawbacks

Aero engine makers were using 1-cylinder rig test engines in the '20s, the object being of course to obtain data, especially on performance, more quickly and cheaply than building and running complete engines.

Rolls-Royce were slower than others to adopt this experimental practice* but in mid-1929 a set of 1-cyl. units was authorised to cover their product range (1016).

____

*Slow on *1-cyl*. engines, but actually Henry Royce had built in 1918 a 60V-2-cylinder rig test engine to prove the novel compound-angle valve gear and fork-and-blade con.-rod system of the new large 60V12 just designed, which became the *Condor*.

R-type 1-cyl. testing

However, R-R recorded that their 1931 1-cyl. rig, representing the Schneider Trophy R-type engine and planned for fuel and sparking-plug tests, proved very unreliable mechanically, giving far more trouble than the same parts in a complete engine (1015 Appendix). Quite likely this was the result of greater vibration.

PV12 1-cyl. testing

In mid-1934 the 1-cyl. rig representing the R-R PV12 (the base of the *Merlin*) was tried with a new cylinder head layout in place of the standard 'flat' head which had 4 valves/cyl. at VIA = 0 and no squish. This new 'Ramp' head had a pair of exhaust valves in line with the cylinder axis and a pair of inlet valves at VIA = 42^0, i.e. inclined inwards and it had some squish. An improvement in performance was observed and so the main engine was redesigned in the same way. On test of this engine, no benefit was found, the heads cracked and exhaust valves failed rapidly. Eventually a 100 Hour Type Test could only be passed with a concession to allow valve changes at part-time (328), although production of this Mark 1 *Merlin* had already been authorised. Because of the problems the General Manager, Ernest Hives (previously head of Experimental for many years), had in the meantime ordered a reversion to the 'flat' head. This remained unaltered in valve layout throughout *Merlin* and enlarged *Griffon* production.
See Fig. N106A.

The 172 Mark 1s produced were fitted in early Fairey *Battle*s (a type which unfortunately proved in 1940 to be fatally useless for its light bombing role when without fighter escort and in the face of German 37 mm and 20 mm *flak*. No blame attached to the engines).

It was decided later that the 1-cyl. performance gain was due to a favourable airflow ramming effect of its inlet tract which was not reproducible in the main engine (901).

The need for caution

The uses of History are "*To inform, inspire – and warn!"*. The above cautionary accounts certainly fulfil the last purpose. It is perhaps not necessary to go as far as Admiral of the Fleet Lord Fisher's dictum "*The best scale for an experiment is 12 inches to a foot!*" ("Memories and Records", 1920). [Those were the days when "*The Royal Navy always travelled 1st class!*".]

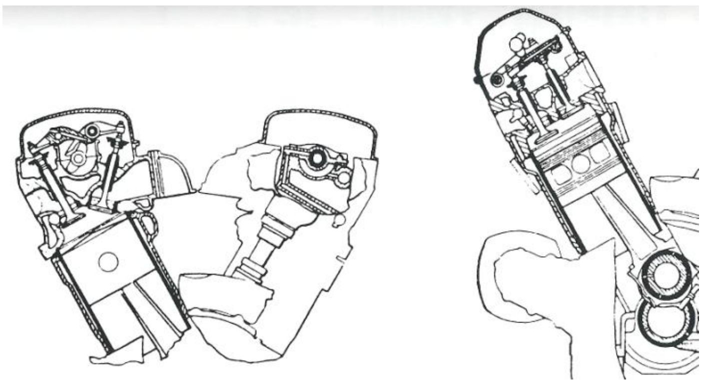


Fig.N 106A

LHS. The "Ramp" head had squish plateaux fore-and-aft of the 4 valves. The inlet valve was inclined at 42^0 to the exhaust to ease the inflow.

RHS. The "Flat" head

DASO 900

Note 107

Probable Inlet Valve Diameter (IVD) for some recent 4 v/c engines
(with a **Sub Note** on Porsche P01)

N.B. IVD is always taken in this review as Overall Head Diameter.

In Appendix 1 there are some recent engines for which Bore (B), Stroke (S) and Peak Power RPM (NP) are known but not IVD. A good estimate of their IVD was needed so that Mean Gas Velocity at inlet at Peak Power (MGVP) could be calculated for comparison with other units.

For this purpose the known IVD and B figures of a wide sample of 4 v/c racing engines over the period 1982 – 2001 was tabled on P.2 and plotted on P.3. The data have been extended subsequently to 2012 engines.

The relation between IVD and B was quite close, with an original linear regression value of IVD/B = 40.6% - the subsequent extension gave the slope shown of 41.1%.

If the combustion chamber does not overhang the cylinder bore, a feature which *was* used in Henri's Peugeots in order to obtain larger valves before it was appreciated that this not only caused "*Negative Squish*" but also *too low* a value of Mean Gas Velocity at inlet (MGVP) (see Note 34), the space available for the inlet valve in a 4 v/c design is 50%B. Therefore an average valve at 41.1%B occupies about 82% of the space available. The balance seems to have been found necessary by many designers for strength of the head around the seats and also to avoid masking the inlet flow near the cylinder wall. Concerning the latter, the 42.1%B obtained in the 2000 Ferrari 049 (Eg. 85) was with a 6^0 longitudinal included angle between the valves, which takes the lifted heads away from the wall. The 1985 Porsche P01 (Eg. 66) had the same feature, although it had the smallest IVD/B at 37.2% (see the Sub Note). A 3-dimensional cam is needed to operate valves with such longitudinal inclination.

Sub Note: Porsche P01

Ref (21) provides the 4 v/c P01 IVD as 30.5 mm in B = 82 mm so IVD/B = 37.2%. This is nearly 4 %points below the relation established from many other engines, which would have given a diameter of 33.7 mm, 10% larger.

This is *assuming* the given dimension as being overall head diameter, the same as the other figures. It is possible that the figure given by Porsche to ref. (21) was an *inlet port throat diameter.* This is really what is needed for a proper scientific analysis and it was often the figure provided for early engines (although adjusted in this review to overall head when the fact was known). It is simply not available in most cases.

Without a cross-section drawing the suggested possibility cannot be checked.

Year	Source	Make	Model	B - mm	IVD - mm	IVD/B %	IVA/PA	VIAX deg.
		CoY Examples						
		Appendix1						
1982	Eg. 62	Cosworth	DFV	85.6742	34.5	40.3	0.324	
1983	Eg. 64	BMW	M12/13	89.2	35.5	39.8	0.317	
1985	Eg.66	Porsche	PO1	82	30.5	37.2	0.277	Some
1988	Eg.71	Honda	RA168E	79	31	39.2	0.308	
2000	Eg. 85	Ferrari	49	96	40.4	42.1	0.354	6
		"Significant Other" Examples						
		Appendix 1						
1992	SO20	Honda	RA122E/B	88	36.5	41.5	0.344	
1995	SO22	Opel	DTM/1*	89	36.1	40.6	0.329	
1995	SO23	Vauxhall	XE20BTCC**	88	35	39.8	0.316	
	DASO		Other					
1983	59,62,554	Cosworth	DFY	90	36.1	40.1	0.322	
1988	47,62	Cosworth	DFR	90	37.1	41.2	0.340	
1989	47,81	Judd	EV	99	40.5	40.9	0.335	
1991	468,	Mercedes	M292	86	36	41.9	0.350	
1996	672,	MugenHonda	MF301HA	93	38	40.9	0.334	
1996	674,	YamahaJudd	OX11A	90	37.5	41.7	0.347	
1996	494,495,	Suzuki	GSXR750	72	30	41.7	0.347	
2001	1039,1040,	Ducati	996R	100	40	40.0	0.320	
1967	G.Beale	Honda	RC174	41	17	41.5	0.344	
2005	1107	Cosworth	TJ	95	41	43.2	0.373	
2005	1095	BMW	P85	98	41.5	42.3	0.359	
2006	1107	Cosworth	CA	98	41.3	42.1	0.355	6
2009	1091	Toyota	RVX-09H	96.8	41	42.4	0.359	3.2
2012	1109	Ducati	999	104	42	40.4	0.326	
2012	'	'	Panigale	112	46.8	41.8	0.349	
2012	'	Kawasaki	ZX-10R	76	31	40.8	0.333	
2012	'	BMW	S1000R	80	33.5	41.9	0.351	

*Cosworth type KC

**Originally Cosworth type KBA in 1987

VIAX = Valve Included Angle in longitudinal direction

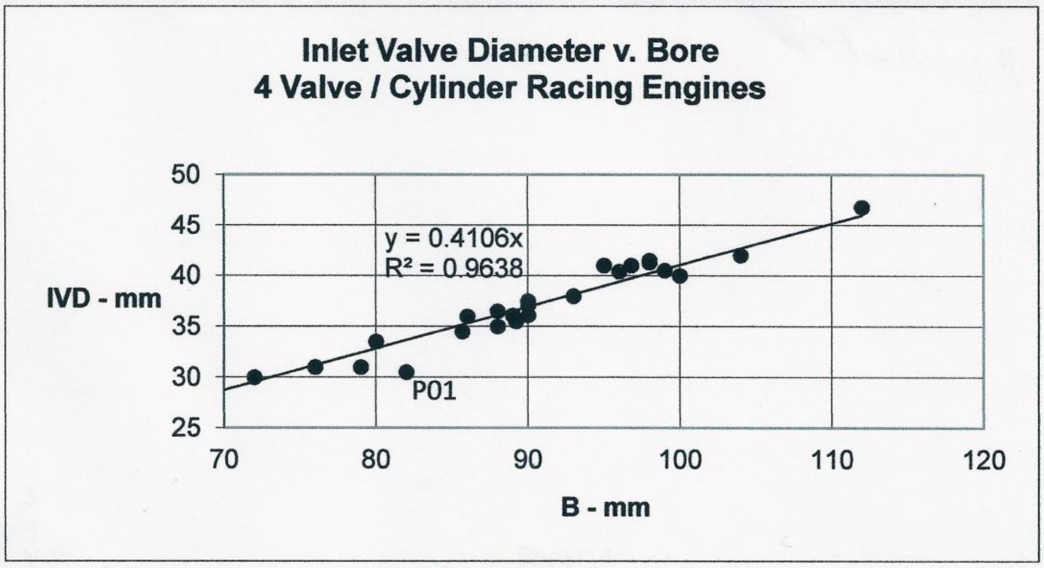
<u>DASO</u> (Additional to Appendix 3)

1091; *Race Engine Technology* No.49, Sept/Oct 2010.

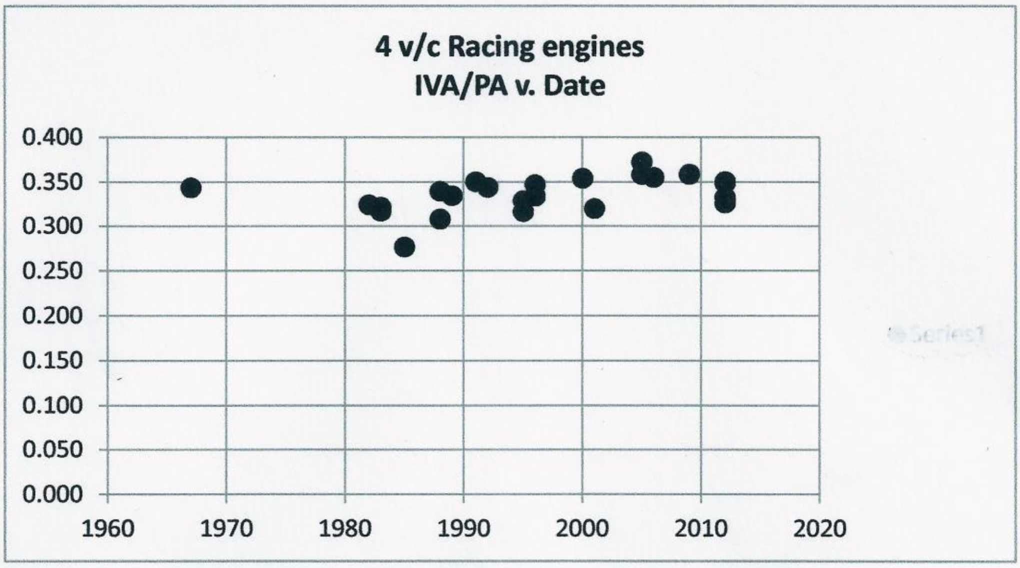
1095; *Ten Years of BMW F1 Engines*; Prof. Dr-Ing. Mario Theissen at al. 2010.

1107; *Race engine Technology* No.73, Sept/Oct 2013.

1109; *Performance Bike* Oct 2012.



When IVD/B = 0.4106, IVA/PA = 0.337.



Note 108

Cosworth 2006 Type CA Series 6   Eg SO25

DASO (1069) and (1070) provided official Cosworth data for this engine and (1092) added details of the engine when re-introduced for 2010 as the type CA2010 (with the FIA RPM limit of 18,000). Further details were given in (1107). Official data are shown underlined.
90 V8 Bore (B) = 98 mm (rule maximum permitted); Stroke = 39.75 mm.
Swept Volume (V) = 2,399 cc (rule maximum 2,400 cc).

B/S = 2.465;
100/Smm = 2.515;

PP = 755 BHP
@ NP = 19,250 RPM.

TP = 214.4 lb.ft.   (1070)
@ NT = 17,000 RPM
$\left(\frac{NP - NT}{NP}\right)$ = 11.7% Fixed Inlet (by FIA rule).

Compression Ratio (R) = 13.3 (1107), so ASE = 0.645.

Inlet Valve Head Diameter (IVD) = 41.3 mm, so IVA/PA = 0.355.
Inlet Valve Lift (IVL) = 16 mm, so IVL/IVD = 0.387 (2006 spec.).
Valve Included Angle (VIA) = 18^0 plus 3^0 longitudinal inclination.

PP/V = 314.7 BHP/Litre;
MPSP = 25.51 m/s;
BMPP = 14.63 Bar;
ECOM (EV x EC x EM) = 59.7%   (Suggested EV = 1.3; EC = 0.7; EM = 0.67).

Weight (W) = 95 kg (rule minimum);
PP/W = 7.95 BHP/kg.

Mean Gas Velocity at inlet @ NP (MGVP) = 71.8 m/s.

BNP = 31.4 m/s.

Con. Rod Length between centres/Stroke (CRL/S) = 2.570
Maximum Piston Deceleration @ NP(MPDP) = 9,834 g.

A Power curve for the CA series 6 was given in DASO (1070) and is shown (redrawn) on P.2.

(1107) states that the CA in 2013 mod. standard, running “full rich” mixture for maximum power, had a Specific Fuel Consumption of 275g/kW.Hr. Allowing for the fuel being, by rule, 94.25% petrol + 5.75% ethanol this was a Brake Thermal Efficiency of 30.0%.
In that year PP was 768 BHP @ 17,250 RPM (BMPP = 16.61 Bar @ 22.86 m/s), corresponding to Volumetric Efficiency (EV) = 1.45. IVL had been increased to 17 mm (IVL/IVD = 0.412), worth 2 HP. Improved fuel and oil had contributed over 8 HP (1107).

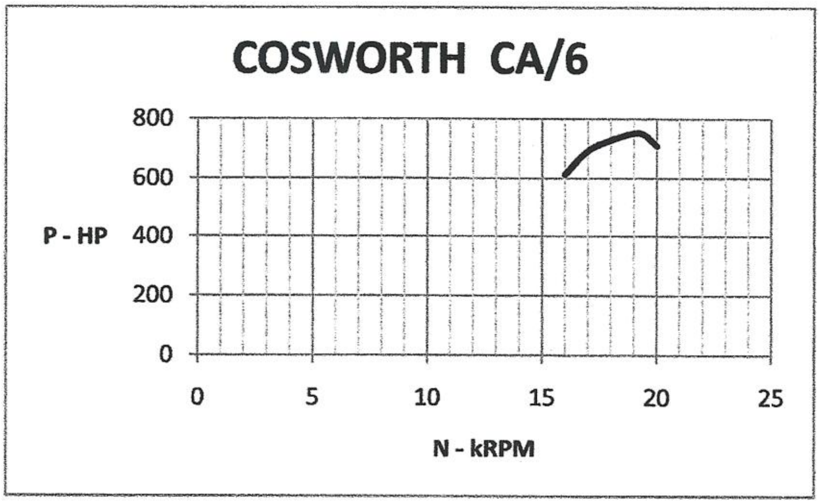
A 2013 engine illustration is shown on P.3.

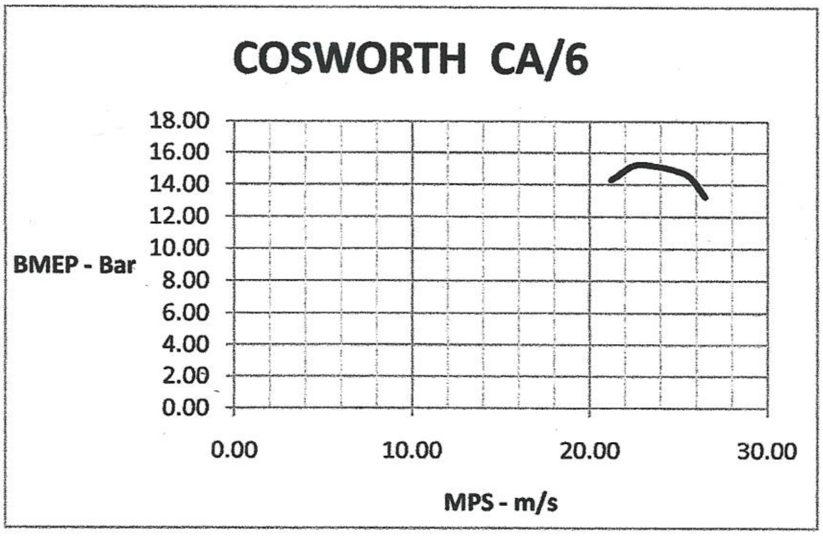
References

(1069) *Race Engine Technology* No. 20 Feb. 2007.
(1070)   “   21 Mar/Apl 2007.
(1092) Cosworth website consulted 6 Oct 2010.
(1107) *Race Engine Technology* No. 73 Sept/Oct 2013 (a particularly comprehensive report).

POWER CURVES

Eg.	SO25			
DASO	1070	and 1092	Note 108	
YEAR	2006			
Make	Cosworth			
Model	CA/6			
Vcc	2399			
Ind. System	NA			
Confign.	90V8			
Bmm	98			
Smm	39.75			
	N	P	MPS	BMEP
	kRPM	HP	m/s	Bar
	16	614	21.20	14.31
	17	694	22.53	15.23
	18	731	23.85	15.15
	18.5	744	24.51	15.00
	18.75	750	24.84	14.92
	19	754	25.18	14.80
	19.25	755	25.51	14.63
	19.5	747	25.84	14.29
	20	711	26.50	13.26





Race Engine Technology No. 073 Sept/Oct 2013   CA2013

In the new 2006 formula the CA powered the Williams FW28 but the relatively low-budget team were uncompetitive, finishing 8th in the Constructors' Championship with only 11 points. The highest finishes were 2 x 6th places.

Cosworth did not compete in 2007 to 2009 but returned in 2010 with Williams (FW32) and 3 low-budget teams (Lotus, HRT, and Virgin). Results were again disappointing, Williams finishing 6th in the Constructors' Championship with a best 4th place. The small teams did not score a point.

The Williams FW33 again used the CA in 2011 but obtained only 5 points for 9th place in the Constructors' Championship. HRT and Virgin also fitted the engine but scored no points. In 2012 the CA powered the Marussia and HRT cars but with no points and in 2013 it was used only by Marussia with no success.

Note 108B

Recent increase in Naturally-Aspirated (NA) Peak Power BMEP (BMPP)

Over the 8 year period of the 2.4 Litre NA formula from 2006 to 2013 a typical Grand Prix engine BMPP rose from about 14 Bar to about 16 Bar. This important increase was stimulated by the imposition by the FIA of limits to the permissible "Red Line" RPM. In 2007 this was 19,000, reduced to 18,000 in 2009

This Note analyses – mostly in qualitative terms – how the increased BMPP was achieved by Cosworth. The basic data was given in Race Engine Technology issue 73 of September/October 2013 (DASO 1107). The power figures are taken from the published chart and vary slightly from a previous source (DASO 1070).

Development of the Cosworth CA series from 2006 to 2013

90V8 Bore (B) 98 mm/Stroke (S) 39.77 = 2.464 Swept volume (V) 2,400 cc

2006 Series 6 Unlimited RPM

Peak Power (PP) 750 BHP @ 19,000 RPM (NP)

Peak Power BMEP (BMPP) = 14.72 Bar @ Peak Power Mean Piston Speed (MPSP) = 25.19 m/s.

2013 Series 19 Max. RPM (Red Line) by rule 18,000

PP 768 BHP @ 17,250 RPM

BMPP = 16.60 Bar @ MPSP = 22.87 m/s.

There was therefore +12.8% BMPP at -9.2% MPSP.

On the CA.6 Power Curve, if the RPM had simply been pulled back to 17,250 RPM the value of PP would have been 700 BHP and BMEP 15.13 Bar (+2.3% above 14.72, which would mostly have been gained from lower friction). So development produced an extra 768 – 700 = 68 BHP (+9.7%).

Identified changes from Series 6 to Series 19.

Grouping the identified changes:-

Retuning to lower RPM

- Longer inlet trumpets
- Revised cams
- Exhaust changes

+ Volumetric Efficiency

- Larger airbox ("snorkel")
- Inlet valve lift increased from 16 mm to 17 = +2 HP

+ Mechanical Efficiency

- Reduced crankcase pressure; depression increased from 125 mBar to 425
- Reduced oil flow
- Lower viscosity oil
- Oil run hotter; +10^{0}C
- Improved oil (Castrol) = +3.5 HP

Other

- Improved BP fuel = +5 HP

Note 109

1991 Engine RPM

Ref. (743) published a list of the RPM of all competing 1991 engines obtained by sound recording and analysis at the finish line on the Estoril circuit during the Portuguese GP meeting. The occasion was the 1st Practice session on 20 September 1991, so the *caveat* is that teams may not have been using all the RPM available. At that place and time it would not be expected that the engines were reaching the "Red line" RPM.

The places obtained in the GP have been added.

All engines 3.5L NA

RPM rank 1st Practice	Engine type	RPM Configuration V8	V10	V12	GP Result
1.	Renault RS3B		14,330		1st
2.	Honda *RA121E/B*			14,051	2nd
3.	Ferrari 037			13,790	3rd
	[Minardi customer Ferrari			n.a.	4th & 9th]
4.	Honda RA101E		13,079		13th
5.	Judd GV		12,925		DNF
6.	Yamaha OX99			12,875	12th
7.	Lamborghini 3512			12,808	11th & 16th
8.	Ford –Cosworth HB5	12,676			5th, 6th, 8th, & 10th
9.	Ilmor 2175A		12,671		7th & 17th
10.	Hart-Cosworth DFR	11,750			15th
11.	Cosworth DFR	11,334			DNF or failed to Qualify

(Judd EV V8 was 14th)

The slowest engine on this occasion was therefore 21% below the fastest. This is accounted for by the difference in Stroke (S), as shown on the following table of Mean Piston Speed (MPS). The various forms of valve gear are identified as:-

Pneumatic Valve Control System: PVRS   Ti-alloy valves: Ti   4 valves/cylinder: 4v/c
Steel Coil Valve Return System: CVRS   Steel-alloy valves: St   5 valves per cylinder: 5v/c

All 3.5LNA; all DOHC & 4v/c; **except** where shown 5v/c

Engine type	Bore (B) mm	Stroke (S))	B/S	N RPM	MPS m/s	BN	Valve gear
Renault RS3B	*93*	*51.5*	*1.806*	14,330	*24.6*	*22.2*	PVRS Ti
Honda *RA121E/B*	86.5	49.6	1.744	14,051	23.2	20.3	PVRS Ti
Ferrari 037	88	47.9	1.837	13,790	22.0	20.2	CVRS Ti 5v/c
Honda RA101E	92	52.55	1.751	13,079	22.9	20.0	*CVRS, Ti**
Judd GV	*94*	*50.4*	*1.865*	12,925	*21.7*	*20.2*	*?*
Yamaha OX99				12,875			CVRS St 5v/c
Lamborghini 3512	87	49	1.776	12,808	20.9	18.6	CVRS St 5v/c
Ford-Cosworth HB5	94	63	1.492	12,676	26.6	19.9	CVRS Ti
Ilmor 2175A	86.6	59.4	1.458	12,671	25.1	18.3	CVRS St
Hart-Cosworth DFR	92.6	64.8	1.429	11,750	25.4	18.1	CVRS St
Cosworth DFR	90	68.6	1.312	11,334	25.9	17.0	CVRS St

*The RA101E customer engine was *assumed* to be *CVRS Ti.*

Average MPS (sample of 10) 23.8 + or – 12%

Valve gear:-Renault, the pioneer of PVRS with 6 years experience, at about 22 m/s in BN shows a 10% advantage from the Honda RA121E/B, both using Ti-alloy valves. The preceding standard, CVRS St, averages just under 18 m/s so the advantage of the new "Top-end" with PVRS Ti was about 24% in BN. The BN gain from Ti valves alone is about 12% (HB5).

Note 110

Notable small mistakes with serious consequences

- 1912 French Grand Prix. Eg. 4. Two-thirds of the FIAT and Peugeot teams suffered broken fuel lines, almost certainly due to their not being sufficiently insulated from 4-cylinder vibrations. One of the FIATs was leading at the time.
- 1924 French Grand Prix. Sunbeam. See under Eg. 10. The team of supercharged cars had the speed to win but were all fitted with new, untested magnetos on the night before the race. These had partial failures which caused misfiring.
- 1936. Velocette 500 cc. Single-cylinder engine vibration caused frothing in the carburetter float chamber and power loss. The motorcycle, ridden by the ace Stanley Woods, lost the Senior TT by 18 seconds in a 3 hour race. Subsequently the float chamber was mounted remotely from the engine.
- 1939. Eg.25. Mercedes-Benz. The supplementary carburetter choke for maximum power had the throttle valve stick open in the French and German GPs, when Hermann Lang was leading. The super-rich mixture resulting at lower RPM ruined the engines.
- 1954. Maserati. The oil tank on this type was moved from the engine bay to the tail to keep the oil cooler. When Stirling Moss was leading the Italian GP one of the long pipes connecting to the rear tank broke because it was badly supported and the engine seized.
- 1955. Eg. 35. Mercedes-Benz. All 3 engines broke at Monaco because a screw in the newly-modified desmodromic valve gear broke in fatigue. Two of the cars retired in succession while leading.
- 1955 – early 1957. Eg. 37. Vanwall. Many retirements were caused by fuel injection pipe or throttle linkage failures in fatigue from 4-cylinder vibrations. The cures were flexible pipes in the injection system and a portion of the same hose inserted into the linkage.
- 1970. Eg. 49. Cosworth. Many failures occurred because an outside supplier ground through the Nitridingon the crank pins into the corner fillets.
- 2000. Ilmor. Failure of a filter on the air pump used to pressurise the pneumatic valve-return system led to 2 engine failures in practice for the Australian GP and 2 more in the race.

Toyota 2009 Type RVX-09 Specn. H: SO26

Toyota announced that it was to compete in the Grand Prix arena in 1999 and made its first entry in 2002. The cars and engines were designed and developed at the company's motor sport base in Cologne. There followed 8 seasons of racing with minimum success, no wins being secured with 4 years of 3L V10 engines and 4 years of 2.4L V8 engines. The firm retired at the end of the 2009 season with best results totalling 5 x 2nd places.

Toyota also supplied engines to Jordan in 2005 (best result 1 x 3rd place), to Midland (ex Jordan) and Spyker in 2006 and to Williams over 2007 to 2009 (best results totalling 2 x 3rd places).

Although not therefore anywhere near "Car of any Year" Toyota did publish a considerable amount of engine data after their retirement in DASO1091 (*Race Engine Technology* No. 49 Sept/Oct 2010). This has been analysed as follows. Official Toyota data is underlined.

90V8 Bore B = 96.8 mm
Stroke S = 40.75 mm*
Swept Volume V = 2,399 cc (Rule maximum 2,400 cc)

B/S = 2.375
100/Smm = 2.454

PP = 772 PS = 761 BHP (deduced from official data)
@ NP = 17,350 RPM (rule maximum = 18,000 RPM)
TP = 231 lb ft (produced by analysis and plotting
@ NT = 16,050 RPM from DASO 1091, p.26.)
$\left(\frac{NP - NT}{NP}\right)$ = F = 7.5% (Inlet tract length (LIN) not variable, by rule)

R = 13.6 on RON102 fuel containing 0.6% Ethanol
ASE = 0.648

IVD = 41 mm (42.4% of B) $\left(\frac{IVA}{PA}\right)$ = 0.359

IVL = 15.4 mm $\left(\frac{IVL}{IVD}\right)$ = 0.376

4 v/c; PVRS; Finger followers; Solid Ti-alloy valves.
VIA = 21.2^0 + 3.2^0 in the plane of valve pairs, i.e., longitudinally

PP/V = 317.2 BHP/Litre
MPSP = 23.57 m/s
BMPP = 16.36 Bar
ECOM = (EV x EC x EM) = $\left(\frac{BMPP}{38 \times ASE}\right)$ = 66.4%
BMTP = 16.4 Bar

W = 95 kg (rule minimum, using some ballast)
PP/W = 8.01 BHP/kg

---

*Official Stroke = 40.77 mm but this gives V = 2,400.3 cc, so Stroke has been adjusted to official 2,399 cc

MGVP = 65.7 m/s

BNP = 28.0 m/s

MVSP assuming IOD = 320^0 = 10.0 m/s

MJ = 44 mm; $\left(\frac{MJL}{MJ}\right) = \frac{18}{44} = 0.41$

CP = 36 mm ; $\left(\frac{CP}{S}\right)$ = 88.3%; $\left(\frac{CPL}{CP}\right) = \frac{17.5}{36} = 0.49$ $\left(\frac{CP\ \%}{S}\right)/\sqrt{(BNP)} = 16.7$

$\left(\frac{CRL}{S}\right) = \frac{111}{40.75} = 2.724$; MPDP = 8,114 g

A Power Curve for the RVX-09H is given on P.3.
A section of the cylinder head is given on P.4.

DASO 1091 also provides data on some of the 90^0V10 3L RVX engines
This is analysed below.

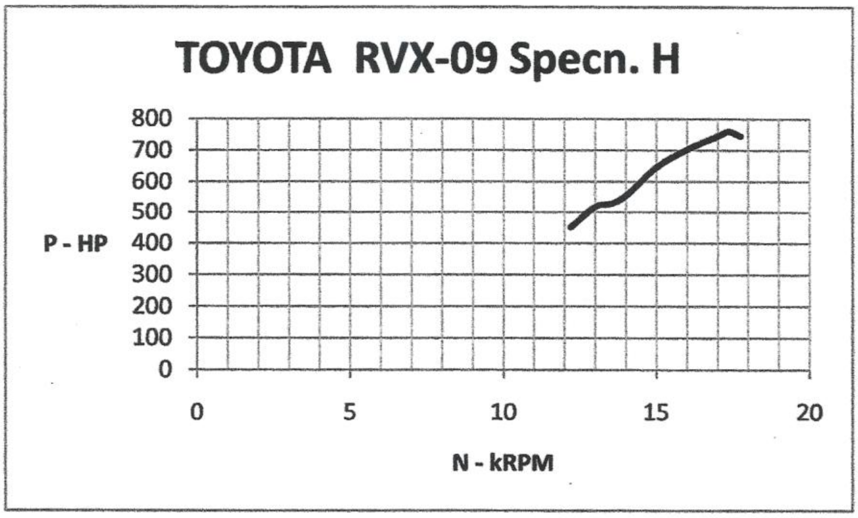
All
90V10
B = 96.8 mm
S = 40.75 mm
V = 2,999 cc

Year	2002	2005	
Specn.	C	F	
PP	848	937	BHP
@ NP	16,850	18,600	RPM
MPSP	22.9	25.3	m/s
BMPP	15.02	15.03	Bar
R not given; assuming 13 then ASE = 0.642			
``and ECOM	61.6%	61.7%	
TP	274.8	284	lb.ft.
@NT	14,600	15,600	RPM
BMTP	15.6	16.1	Bar
F	13.4%	16.1%	LIN variable for both.
BNP	27.2	30.0	m/s
W		Less than 95	kg

POWER CURVES

Eg.	SO26			
DASO	1091			
YEAR	2009			
Make	TOYOTA			
Model	RVX-09 Specn. H			
Vcc	2399			
Ind. System	NA			
Confign.	90V8			
Bmm	96.8			
Smm	40.75			
	N	P	MPS	BMEP
	kRPM	HP	m/s	Bar
	12.2	452	16.57	13.82
	13	518	17.66	14.86
	13.5	528	18.34	14.59
	14	555	19.02	14.79
	15	647	20.38	16.09
	16.05	706	21.80	16.41
	17	746	23.09	16.37
	17.35	761	23.57	16.36
	17.75	746	24.11	15.68

Powers as published were German PS and have been divided by 1.014 to convert to HP



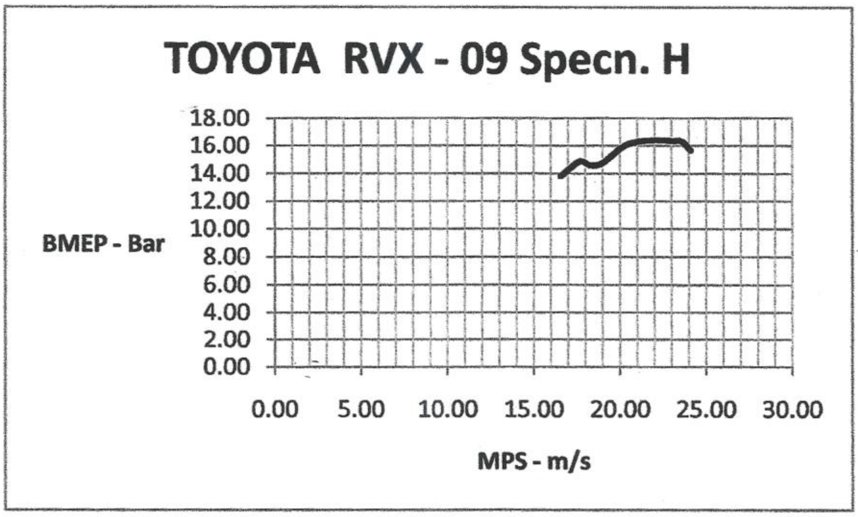


Fig. SO26A

2009 Toyota RVX-09 Specn. H

90V8 96.8/40.75 = 2.375 2,399 cc

Note the large diameter, thin-walled camshafts; although suggesting a built-up construction the shafts and the (heavily-relieved) cams were actually 1-piece in accordance with FIA F1 rules.

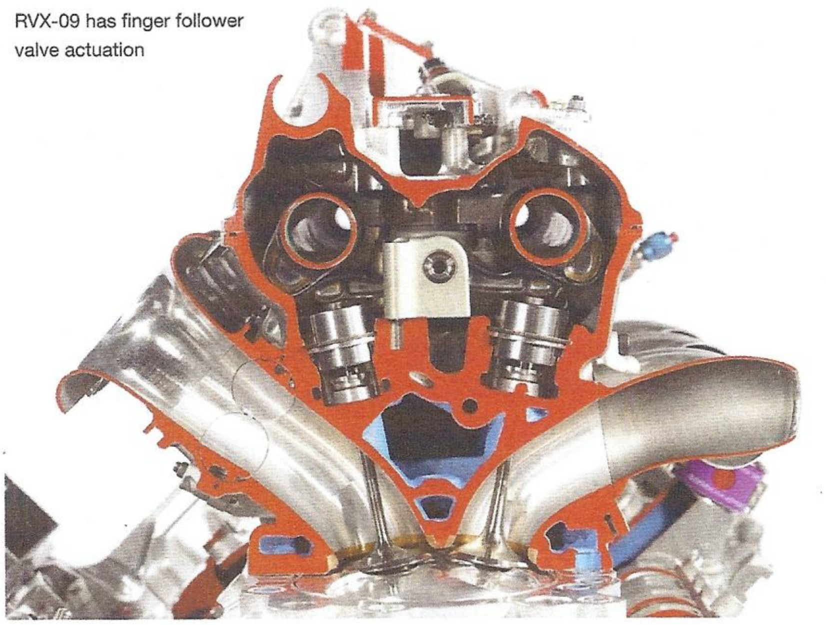
Finger followers and PVRS.

All rubbing surfaces were DLC coated as well as lubricated.

The camshaft driving gears contained an 8-roller pendulum damper.

VIA = 10.2^0 inlet + 11^0 exhaust = 21.2^0, with 3.2^0 between valve pairs in the fore-and-aft plane.

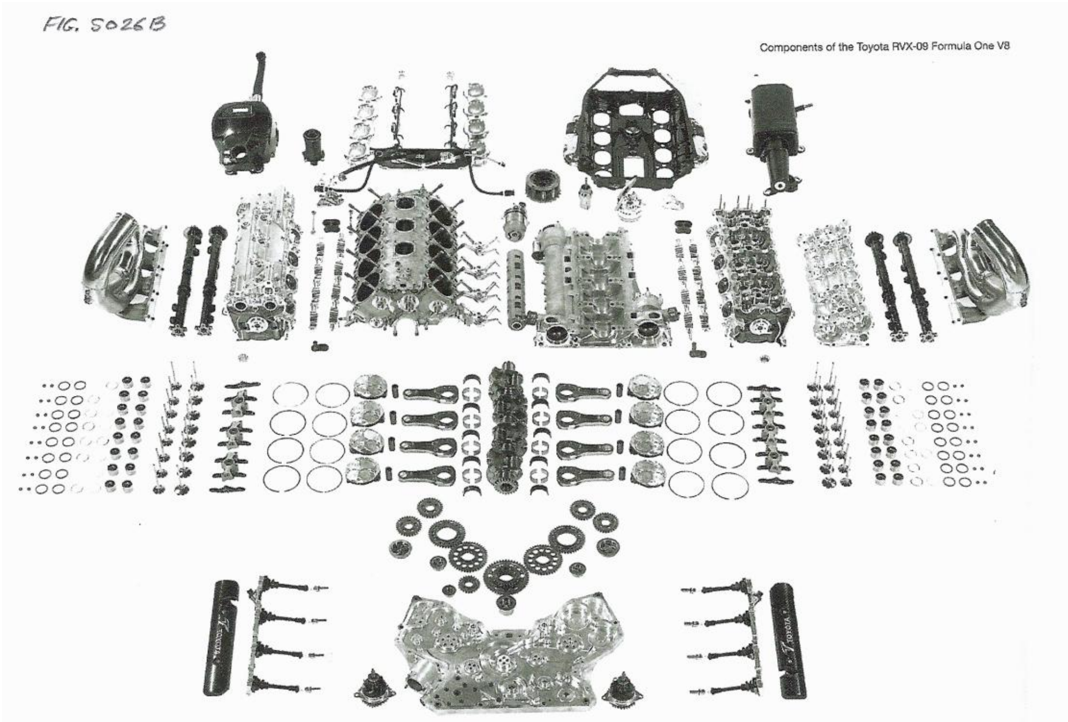
The inlet tract is shaped to provide in-cylinder "Barrel Turbulence" (aka " Tumble Swirl").



DASO 1091

Fig. SO26B

Components of the RVX-09/H



DASO 1091

P.S.
Some thoughts on the lack of success of the Toyota Grand Prix campaign

The official Toyota data for the RVX-09H shows that its power was on a par with, eg the Cosworth CA (see Note 108). Nevertheless, a former senior Cosworth engineer closely associated with the CA told the author in 2010 that he thought the RVX was about 30 HP *less* powerful.

This opinion must have been judging the poor Toyota results as stemming from the engine alone. If this, on the evidence of the data released shortly afterwards, was *not* the case, then the chassis and/or the drivers must be identified as "equivalent" to 30HP deficiency.

Success breeds success *and vice versa* where availability to a team of front-rank drivers is concerned. Toyota over 2002 to 2009 did not have the services of Michael Schumacher, or Alonso, or Raikkonen, or Hamilton or Button – to name the Champions of 2002-2004, 2005-2006, 2007, 2008 and 2009.

Ferrari engineers certainly did not think that the Toyota lack of success was due to the engine. Luca Marmorini, an ex-Ferrari man, oversaw the 3L V10 and 2.4L V8 developments from the start in 1999 to January 2009. By then the RVX-09H was specified and also very probably the Toyota board had by then decided to quit at the end of the season. In October 2009 Marmorini was invited to take the place of Gilles Simon as head of Ferrari's Engine & Electronics department - a large vote of confidence.

It may be that Toyota provided the 2009 engine data to show that it was *not* deficient and so justify their knowledge of basic engine technology. Deducing that perhaps chassis technology *was* lacking in the arcane F1 area – quite unrelated to road cars - would not reflect on the firm's basic normal production.

Note 112

BMW 2005 P85: SO27

DASO 1095* provides official BMW data (underlined) for this prototype engine.
It was not raced in the 2005 season because the FIA suddenly changed the required engine life limits from 1 race weekend, to which the P85 had been designed, to 2 race weekends. BMW assessed this as requiring a 1,600 km life instead of 800.

The P85 was the last all-new BMW V10.

4 v/c DOHC
90V10 Bore (B) = 98 mm
Stroke (S) = 39.75 mm
Swept Volume (V)= 2,998.33 cc
B/S= 2.465
100/Smm=2.516

Peak Power (PP)=>937 BHP [BMW >950 PS]
@ NP=19,300 RPM [BMW Red Line 19,800]
Assessed 500 below Red Line

Peak Torque (TP)= 265.5 lb.ft. [BMW 360 Nm]

PP/V=312.5 BHP/Litre [using 937 BHP]
BMPP = 14.49 Bar
@ MPSP = 25.57 m/s [using 19,300 RPM]

BMTP = 15.09 Bar

BNP = 31.52 m/s

Inlet Valve head Diameter (IVD) = 41.5 mm 42.35% of B
Inlet Valve Area/Piston area (IVA/PA) = 0.359
Mean inlet Gas Velocity @ PP (MGVP) = 71.29 m/s

Exhaust Valve head Diameter (EVD) = 34.4 mm 83.9% of IVD

Weight (W) = 82 kg
PP/W = 11.43 BHP/kg

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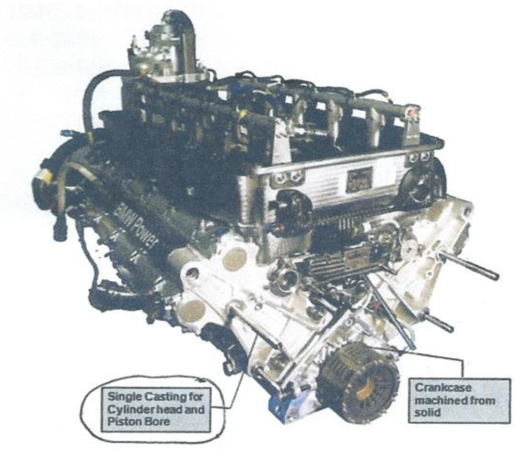
* DASO1095. *Ten Years of BMW F1 Engines*. Paper by Prof. Dr-Ing. Mario Theissen et al. 2010.



P85

Type	V10-90°
Displacement	2,998.5 cc
Bore	98 mm
Stroke	39.75 mm
Cylinder spacing	102 mm
Bank offset	18mm
Engine length	575.0 mm
Engine width	517.0 mm
Engine height	290.0 mm
Weight	82 kg
Centre of gravity height	110 mm
Max. output	>950 PS
Max. torque	360 Nm
Max. engine speed	19,800 rpm
No of valves	40
Intake valves	41,50 mm, titanium
Exhaust valves	34.40 mm titanium

Note 113

The growth of budgets

The effect on the budgets of Grand Prix motor-racing teams of ever-wider TV coverage, inducing more non-motor-industry sponsors to pay increasing amounts for on-car advertising, can be seen from the following comparisons (this funding source opened in 1968 with Lotus' contract with Imperial Tobacco).

Date	Data Source	Team	Budget £M	No. of Races	Retail Price Index	Budget 2002 £M	Results Position in Championships Drivers'	Constructors'
1969	(878)	BRM	0.1	11	63.7	**1**	11th	5th
1983	(878)	Lotus	5	15	335.1	**10**	12th	7th
1999	(879)	Jordan	25	16	652.5	**27**	3rd	3rd
2002	(758)	Ferrari	95	17	695.1	**95**	1st	1st.

Notes on the reliability of Data Sources
(878). *Grand Prix International* 8 June 1983. This gave an interview with Tony Rudd, who was Chief Engineer and Racing Manager of BRM in early 1969 and held the same posts with Lotus Cars/Team Lotus in early 1983, so his budget figures can be relied upon.
(879). A Jordan spokesman quoted their 1999 Budget as $40M in *Autocar* (24?) January 2003, equivalent to £25M at the then-prevailing rate of $1.6/£1.
(758). Figure is an *estimate* in *Autocar* (22?) February 2002 and therefore is not very reliable. The Ferrari figure in that source was exceptional: McLaren-Mercedes were estimated as £85M; Williams-BMW as £70M. The slowest team, Minardi, were put at £30M. The Jordan spokesman in (879) illustrated the growth of costs in only 4 years by contrasting their 3rd place results in 1999 with a rather higher spend by Minardi in 2002 producing only back-marker places.

The BRM and Lotus budgets did not produce the desired results and it must be assumed that the Championship-winning teams spent more (Matra-Tyrrell in 1969; Brabham & Ferrari in 1983).

Scope of budgets

There is doubt over the scope of the budgets listed – did BRM include engine supply? It seems unlikely that it included engine development. Did the figures include the drivers' salaries? In the case of Jordan in 1999 engines *may* have been supplied free-of-charge by Mugen. The Ferrari figure for 2002 *may* have included engine development and *presumably* included the reputed-to-be-extremely-high salary of Michael Schumacher (which was suggested to be £20M in 2001 by (739)).

Team profit

The 1969 BRM budget would have been funded mostly by its owners, the Owen Group, with some help from motor industry contributions. The later sponsored team budgets, with some income from the "circus", may or may not represent income or costs but there was some profit to the team.

Conclusion

Although the precise ratio of fund escalation cannot be certain it is clear that – in constant money value terms - the increase in the money available for Grand Prix technology improvement from 1968 to 2002 was truly remarkable.

Note 119

Opel 1992 Formula 3 Eg SO28

As an early stepping-stone for drivers aspiring to join a Grand Prix team, Formula 3 is designed to be both relatively cheap and not too powerful. Therefore the F3 regulations current in 1992 specified (mainly):-

- A 4-stroke 4 cylinder 2 litre production petrol engine base, retaining the original block and head;
- Natural aspiration (NA) with breathing restricted through a 24 mm diameter, 3 mm long, orifice.

Grand Prix engines have never been breathing-restricted, although it is a common rule in top-level sports car racing and also for NASCAR races on super-speedways. However, the method has interesting aspects which were thought worth examining on this site.

Comprehensive data on the Opel 1992 F3 engine, which powered the German F3 series Champion of that year – Pedro Lamy in a Reynard 923 chassis – were published in Motortechnische Zeitschrift 54 (1993) (DASO 1119). This has been provided to the author recently by the courtesy of Bosse Skånhed. The Opel engine was developed in conjunction with the German tuning firm of Siegfried Spiess, which had previously produced a successful Volkswagen-based F3 engine.

Opel base engine

Opel chose a basic production engine in sporting tune which had the following specification:-

DOHC; 4 v/c @ VIA = 46^0; cast iron block; Al-alloy head;

This engine had been developed with input from Cosworth on the head for Vauxhall (see Appendix 1 Eg SO23 and also Significant Other SO23). It had Barrel Turbulence (aka Tumble Swirl).

IL4 Bore (B) 86 mm Stroke (S) 86 mm B/S = 1 Swept Volume (V) = 1,998 cc.

R = 10.5 so ASE = 0.61

PP = 110 kW = 147 BHP

@ NP = 6,000 RPM

BMPP = 11.06 Bar

MPSP = 17.20 m/s

ECOM = 47.7%

Obviously, the life between overhauls of this series production engine was measured in years, helped by its low Load Factor in typical road service.

Opel F3 engine

The breathing restriction

The inlet restriction of 24 mm diameter means that an efficient 2 litre 4-stroke piston engine can induce enough airflow to "choke" the orifice. That is to say, at a certain RPM the accelerated flow of air reaches sonic velocity in the orifice. After that, at fixed ambient temperature and pressure, there cannot be any increase in velocity or air mass flow.

The choked flow condition is given by:-

- $\left(V/\sqrt{T}\right) = 59.5$; where V = Velocity in the orifice ft/sec;
- $\left(\dfrac{M \times \sqrt{T}}{A \times P}\right) = 0.394$ T = Absolute Ambient Temperature before acceleration ^{0}K

M = air Mass Flow lb/sec;

A = Area of the orifice sq. in.;

P = Ambient Total-head Pressure before acceleration psi.

(Data from DASO 429 *The Compressible Flow of Fluids in Ducts*
R. Jamison & D. Mordell, Rolls-Royce Ltd, 1944).

At ambient Standard Temperature and Pressure (STP) of 288^0K and 14.7 psi (1013 milliBar) and A = 0.7 sq. in. (4.52 sq. cm.) for a 24 mm orifice:-

- V = 1010 ft/sec = 308 m/s;
- M = 0.24 lb/sec = 0.11 kg/sec.

These figures assume negligible friction loss up to the orifice which, with a well-shaped bell entry, is near enough true. What is essential is that the throat velocity is diffused back to a low value, recovering as much as possible of the static pressure, before entry into the engine. The geometry used in the Opel engine is shown on Figs. SO28A & B below (from DASO 1119).

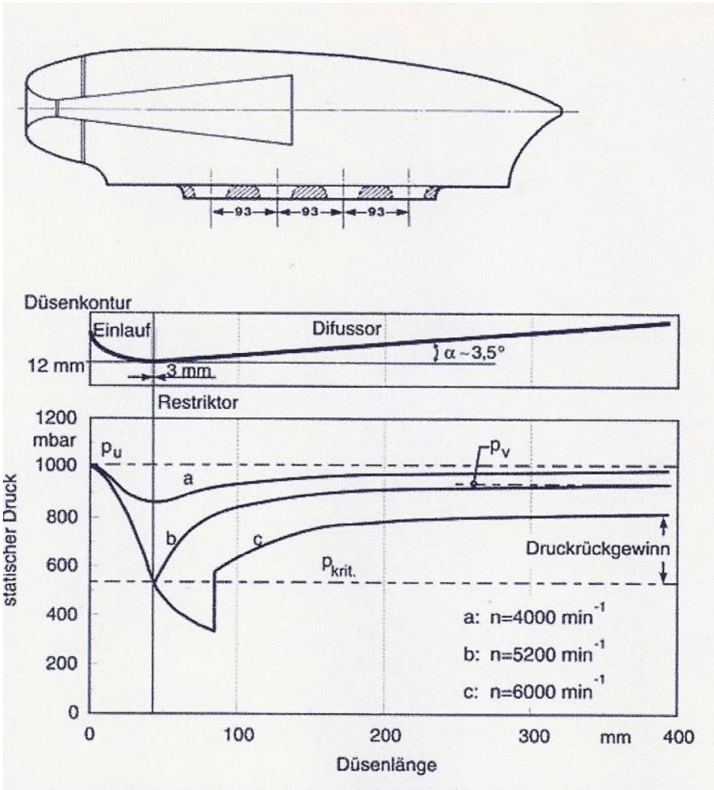


Fig. SO28A
The airbox containing the regulation orifice is mounted externally on the side of the engine cowling and is streamlined to reduce drag.

Fig. SO28B
The upper diagram shows the critical figures for the diffuser:-
Total included angle 7^0;
Area ratio = Exit/Orifice = 7
The lower diagram shows the Static Pressure through the system.
Dusen = Nozzle; *Druck* = Pressure; *Druckruckgewinn* = Pressure recovery.

Line b @ 5,200 RPM is the pressure when the orifice is just choked, when the pressure lost is (p_u - p_v). (NB. The entry pressure should show 1013 mBar).

Line c @ 6,000 RPM shows how the further suction from the engine accelerates the flow from the orifice at a velocity increasing from sonic and then goes through a shock wave of pressure rise to subsonic and steady diffusion thereafter. The air mass flow is unaltered from the value at 5,000 RPM and the pressure loss is greater so power is reduced (see the Power Curve on P. 5).

An engine cross-section is given on P.3

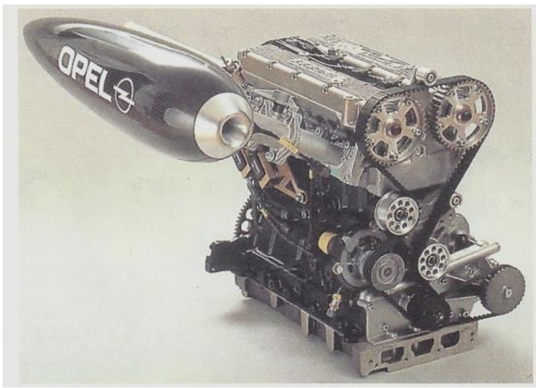


Fig SO28C
Opel F3 engine
The exhaust system, not shown, was 4-into-1. A silencer was fitted to meet a rule noise limit of 106 dBa at 4,000 RPM.

DASO 1119

Fig. SO28D
1992 Opel Formula 3
IL4 86/86 =1 1,998 cc

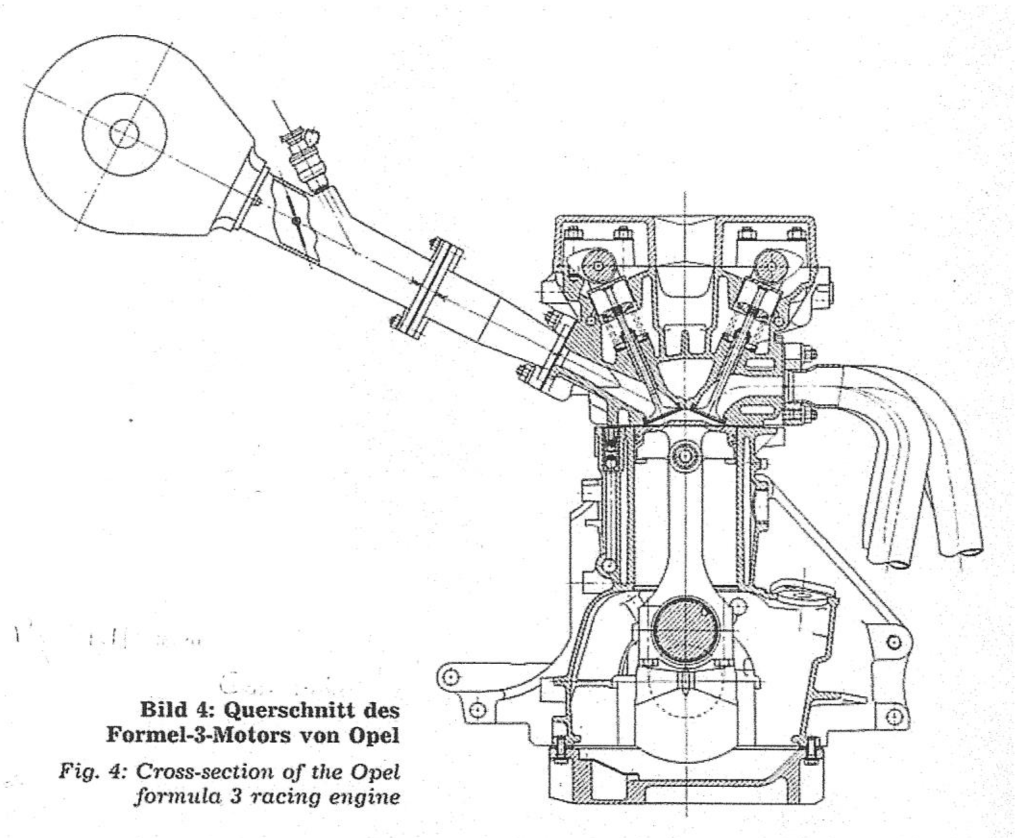


Bild 4: Querschnitt des Formel-3-Motors von Opel
Fig. 4: Cross-section of the Opel formula 3 racing engine

DASO 1119

Performance modifications to basis engine

Long individual inlet tracts were required to tune them to the relatively low speed of the breathing-restricted F3 engine. At 400 mm their length (LIN) corresponded to a resonance at 6,500 RPM on simple theory (see Note 27). This was well above the operating range, with 6,000 maximum, but the airbox probably affected the result. [In 1993 variable length adjustment was fitted, which raised BMEP by 5% at 4,200 RPM].

Valve sizes were restricted by rule to the same sizes as the basis:-

IVD = 33 mm, so IVA/PA = 0.294; EVD = 29 mm.

Camshaft alterations were permitted, so lift was increased:-

IVL = EVL = 11.15 mm (+ 17.4% above basis);
so IVL/IVD = 0.338; EVL/EVD = 0.384.

Valve timing was extended:-

Inlet		Exhaust		
Opens	Closes	Opens	Closes	Rel. to basis
33^0	53^0	58^0	28^0	
early	late	early	late	
IOD = 266^0		EOD = 266^0		$+14^0$ inlet; $+6^0$ exhaust
	OL = 61^0			$+9^0$

These relatively modest timing also reflect the low F3 RPM.

Compression ratio (R) was raised to 12.8 for the unleaded 98RON petrol of the German F3 rules (104RON was allowed in other F3 series), so ASE = 0.639. [In 1993 knock-sensors and ignition adjustment were fitted].

Performance

	Rel. To basis
PP = 129 kW = 173 BHP	+17.3%
@ NP = 5,000 RPM	-1,000 RPM
BMPP = 15.50 Bar	+ 40.1%
@ MPSP = 14.33 m/s	
ECOM = 63.8%	+16.1%points
TP = 256 Nm = 189 lb.ft.	
@ NT = 4,600 RPM	
BMTP = 16.12 Bar	+30.8%
@ MPST = 13.19 m/s	
$\left(\frac{NP - NT}{NP}\right)$ = F = 8%	-12%points

Opel claimed at the time that the value of BMTP at 16.1 was the best known for a Naturally-Aspirated engine. It would have been helped by lower friction losses at the low RPM.

Thermal Efficiency

The Specific Fuel Consumption (SFC) at Peak Power was:-

271 g/kW.Hr (0.446 lb/BHP.Hr)

On petrol this was equivalent to Brake Thermal Efficiency (BThE) = 29.8%.

Volumetric Efficiency

With the given data and by the method described in Note 37 Volumetric Efficiency (EV) is calculated as EV = 1.37.

Other non-critical data

MGVP = 48.66 m/s

This suggests that, being so far below an optimum of around 70 m/s, the performance could have been improved by fitting smaller valves, but this probably was not allowed by the rules.

MVSP = 2.52 m/s

CRL/S = 1.72; MPDP = 1,551 g

The con. rods were 5 mm longer than in the basis engine as a shorter, slipper, piston was fitted (PH/B = 0.42). Rod section was changed from the usual –**I-** section to a proprietary **–H-,** for no obvious reason**.**

All these factors were very moderate because of the low speed.

MJ = 58 mm; CP = 49 mm; GP = 18 mm (reduced from 21 with lighter piston).

Weight

W = 96 kg; PP/W = 1.80 HP/kg.

Weight had been reduced by 34 kg from the basis engine, mostly by discarding un-needed accessories e.g. the generator, but also by certain specification changes (kg):-
4.53 with a lighter flywheel of greatly reduced inertia; 4.23 lighter crank with reduced counter-weighting; 2.2 by cutting off unwanted block mountings; 0.64 lighter (Mg-alloy) camshaft cover; 0.564 lighter rods (total); 0.472 lighter pistons & pins (total); altogether saving 12.6 kg.

Comparison with Basic and Racing-Touring Performance

The Opel 1992 F3 breathing-restricted engine was 17.3% more powerful than its basis engine (173 to 147 HP) but 41.4% less powerful than the same engine modified (as a Vauxhall) for the 1995 British Racing Touring class (173 to 295 HP with an 8,500 RPM max rule limit) (see Appendix 1 SO23 and Significant Other SO23).

Life between overhauls (LBO)

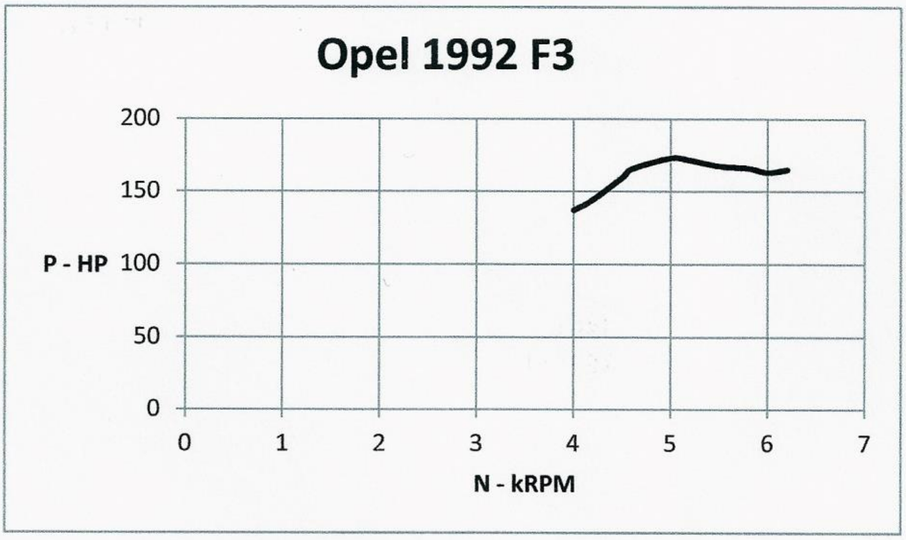
It is believed that the LBO was 1000 racing km.

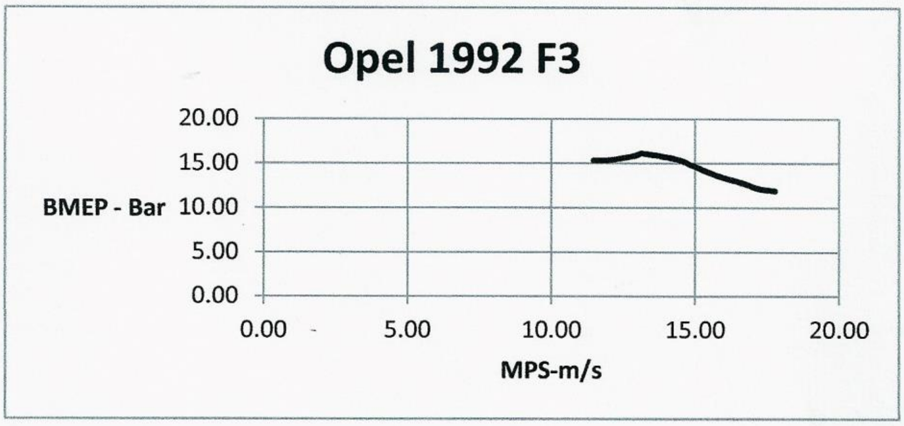
POWER CURVES

Eg.	SO28
DASO	1119
YEAR	1992
Make	Opel
Model	F3
Vcc	1998
Ind. System	NA
Confign.	IL4
Bmm	86
Smm	86

N kRPM	P HP	MPS m/s	BMEP Bar
4	137	11.47	15.34
4.18	143.4	11.98	15.37
4.5	159.5	12.90	15.88
4.6	165.6	13.19	16.12
5	173	14.33	15.50
5.2	171.5	14.91	14.77
5.5	167.5	15.77	13.64
5.8	166	16.63	12.82
6	163	17.20	12.17
6.2	165	17.77	11.92

Powers as published were kW and have been multiplied by 1.3403 to convert to BHP



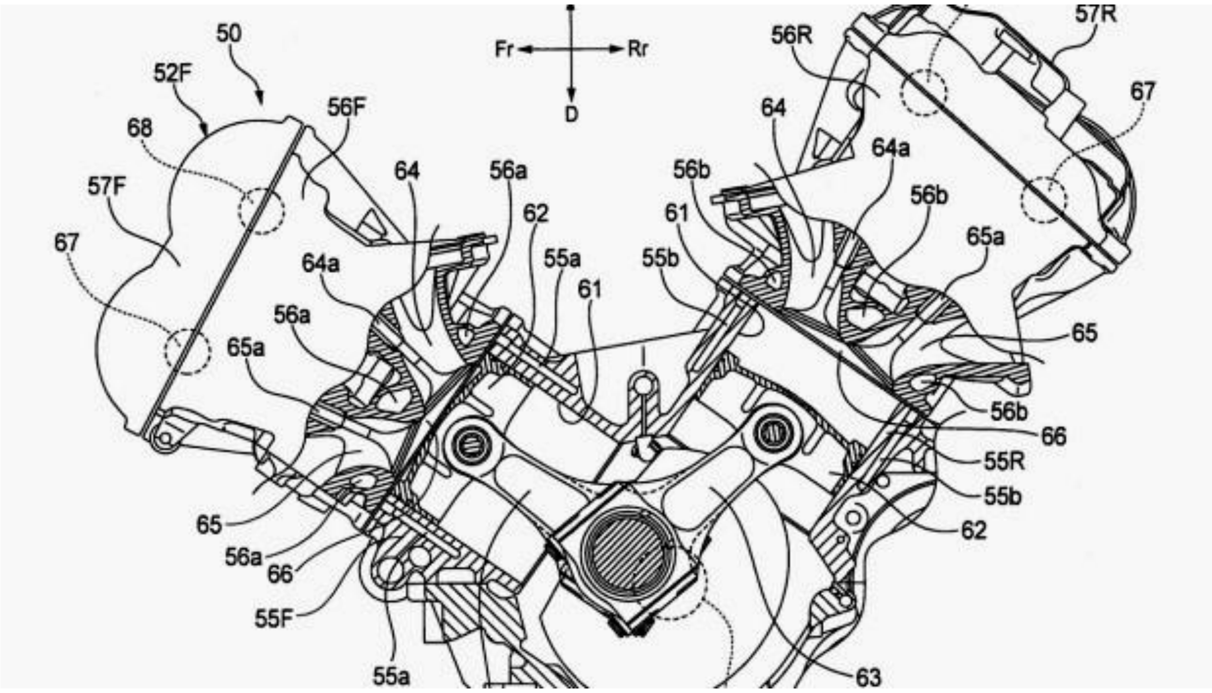


Honda RCV1000RR

A rare cross-section of a modern high-power racing engine is available at:-
http://gas2.org/2014/04/05/honda-rcv-1000-street-engine-revealed/
This is a Honda patent drawing for a motorcycle engine from March 2012. It has been suggested that it represents the 2015 Honda RCV1000RR, intended to be street-legal but also eligible to compete in the "Open" class of MotoGP (i.e. with independent teams, alongside the "Factory" class machines).

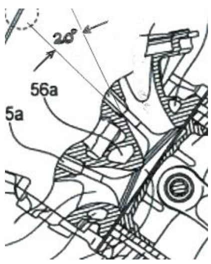
It seems likely from the date that the design shown is based on the 2012 Honda RCV213V factory engine and therefore it gives the latest known internal details of a front-rank racing engine (Honda powered the riders who were 2nd and 3rd in the 2012 Championship, which was the 1st year when MotoGP went to 1,000cc/4 cylinders).

90^{0}V4; Bore (B) = 81mm (rule maximum allowed);
Stroke (S) = 48.5mm; B/S = 1.67.
Swept Volume (V) = 999.7cc.;
Watercooled; DOHC; 4 v/c.
Forward pair of cylinders at 55^{0} to vertical; rear pair at 35^{0}



Various interesting details can be obtained from the drawing, some by scaling (which has a possible accuracy of ∓ ½% from an enlarged print).

- Angle between valves (VIA) = 20^{0};
- Inlet ports are shaped to provide "Barrel Turbulence" (aka "Tumble Swirl"; see The Unique Cosworth Story and Note26) with the outer wall 20^{0} non-orthogonal to the valve head. This is shown in the scrap section at RHS;



- Inlet valve head diameter (IVD) = 31.5 mm;
  so Inlet Valve Area/Piston Area (IVA/PA) = 0.30.
- Inlet port downdraught at head entry = 40^{0};
  exhaust port updraught at head exit = 40^{0}. Relative to plane of cylinder bore.

Mechanical details

- Cylinder barrels are un-lined, i.e. presumably Al-alloy with surface coating;
- 2-ring pistons; Piston Height (PH) = 24 mm. PH/B = 03; PH/S = 0.5.
- Pistons are oil-cooled by spray jets;
- Con-rod Length between centres (CRL) = 89 mm; CRL/S = 1.84. **I**-section**.**
- Plain bearings;
- Crank-pin diameter (CP) = 33 mm; CP/S = 0.68.
- Gudgeon pin diameter (GP) = 15 mm; GP/CP = 0.45.
- The valve operating gear is not sectioned, but it was originally fitted with steel coil springs (Coil Valve Return System (CVRS)).
  It is reported that for Open class MotoGP racing in 2015 a Pneumatic Valve Return System (PVRS) will be fitted, as used on the Factory Hondas. This would be unsuitable for a street engine, since the system has to be pressurised before starting-up.
  More discussion of valve return systems can be found at
  Grand Prix Motorcycle Engine Development, 1949 – 2008 at pp 19 and 20.

Performance

For the 2014 version Honda claimed:-

175 kW (234.5 BHP) @ 16,000 RPM;
therefore:-
BMPP = 13.1 Bar @ MPSP = 25.9 m/s.
MGVP = 86.3 m/s.
BNP = 21.6 m/s.

The RPM for this CVRS engine looks too high. The Factory Honda RC213V in 2012, which had PVRS, was shown by TV at Indianapolis to be reaching a maximum of 15,000. Perhaps the figure claimed for the RCV1000RR was a pre-test estimate. The performance of the machine in 2014 was reported to be below expectations.

If the power-peak RPM was really ,say, 14,000 then the analysed parameters would be:-

MPSP = 22.6 m/s;
MGVP = 75.4 m/s;
BNP = 18.9 m/s.

Without more details it cannot be determined whether the power of the RCV1000RR is limited by the Open class rule fuel ration of 24 litres (100 Octane No. un-leaded) for the usual race distance of around 120 km.

Price of complete Motorcycle.

It is reported that the complete motorcycle price is $100,000. This is with a normal gearbox (6 speeds, limited by rule). The “seamless-shift” gearbox of the Factory Hondas would greatly increase the price of the machine.

Note 121

Honda "Third Era" engines, 2000 – 2008

After leaving the Grand Prix arena in December 2008, Honda published a great deal of information about their "Third Era" engines and their customer and works cars as raced over the seasons 2000 – 2008. This was in a Honda R&D website as "Technical Review: F1 Special" (DASO 1121). This can be accessed after registration. It was brought to the author's attention by a correspondent, Ron Rex, for which I am very grateful. There are 50 papers in this Review, totalling 316 pages.

This Note adds some analysis not given on the Honda site.

Honda began to build Formula 2 and Formula 1 engines in a period lasting from 1964 to the end of 1968, and this they now refer to as "First Era". Some details of these engines have been given on this site in the introduction to the 2nd Pressure-Charged Era (2PC) Egs 69,70,71 plus Additions of November 2013. This was followed by Honda with a further F2 era from 1980 to 1984, which overlapped the 2PC period. More details of this F2 engine are given in Note 94.

Honda refer to their TurboCharged (TC) engines of 1983 – 1988 followed by their Naturally-Aspirated (NA) engines of 1989 – 1992 as "Second Era". Details for the Honda engines which powered the "Grand Prix Car-of-the-Year" in 1986 – 1988 are given in the chapter on 2PC quoted above and of the engines powering CoY in 1989 – 1991 in 3rd Naturally-Aspirated Era (3NA) 1989 -2000. Their 1992 engine is described as SO20 in Significant Other.

In the years 1993 – 1999 Honda kept a "watching brief" on the Grand Prix scene via a separate-but-associated company, Mugen Honda, founded and run by a son of Soichiro Honda. This supplied F1 and F3000 engines to a variety of teams. Their final GP engine of 2000 (used by Jordan and which overlapped the new works Honda engine) was (DASO 1121):-

72 V10 3L; Bore (B) 94.4 mm; Stroke (S) *42.8*;* B/S = *2.2*; Swept Volume (V) *2,996* cc.
Peak Power (PP) *757* BHP; Weight (W) 122 kg; PP/W = 6.2 BHP/kg.

---

*Some data in this Note has had to be deduced or scaled from the Honda review and this is shown in *italics*. This convention is not carried over to the analyses.

During the remainder of the 3L NA formula the works Honda engines were supplied to British American Racing (BAR) over 2000 – 2003, then to a jointly-owned BAR-Honda team 2004 – 2005 before becoming a full Honda F1 team for the 2.4L NA formula 2006 – 2008.

Honda 3L V10 engines, 2000 – 2005

Year	2000	2001	2002	2003	2004	2005
Type	RA000E	RA001E	RA002E	RA003E	RA004E	RA005E
Confign.	80V10	80V10	94V10	90V10	90V10	90V10
B mm	95	95	97	97	97	97
S mm	42.24	42.24	40.52	40.52	40.52	40.52
B/S	2.249	2.249	2.394	2.394	2.394	2.394
V cc	2,994	2,994	2,994	2,994	2,994	2,994
Life Required – km	**400**	**400**	**400**	**420**	**800**	**1500**
[*PP* derived from Fig. 1 of paper F1-SP2_02e; *NP* from Fig. 1 of F1-SP2_06e]						
PP - BHP	*757*	*788*	*859*	*897*	*926*	*938*
@ NP - RPM	*16,500*	*16,750*	*17,500*	*18,250*	18,700	18,700
[Analysis]						
BMPP – Bar	13.71	14.06	14.67	14.69	14.80	14.99
@ MPSP – m/s	23.23	23.58	23.64	24.65	25.26	25.25
[W from Table 1 of F1-SP2_07e]						
W – kg	112	108	111	99	91	89
[Analysis]						
PP/W - BHP/kg	6.76	7.3	7.74	9.06	10.18	10.54
Datum	Datum	+8%	+14.5%	+34%	+50.6%	+55.9%

Some further details of the 2005 RA005E are added here because this was the last Honda engine built before the new 2006 2.4L formula began with very large restrictions imposed by the FIA on what a designer could do (ostensibly to reduce cost – **after** the heavy expense of the new engines! In the author's opinion it was also part of a move long desired by the rule-making body to impose a standard engine and convert F1 racing into simply a Drivers' Championship without troublesome technical interference – like all the junior formulae).

SO29:-2005 Honda RA005E [See Short Glossary of Abbreviations on website]

IVD mm = 41.6; IVD/B = 42.9% (see Note 107 for comparison);
IVA/PA = 0.368; MGVP = 68.6 m/s;
IVL = *13.5* mm; IVL/IVD = 0.325; IOD = *316*^{0}; MVSP = 9.59 m/s.
R = *13*; ASE = 0.642; ECOM = $\left(\frac{BMPP}{38 \times ASE}\right)$ = 61.4%.

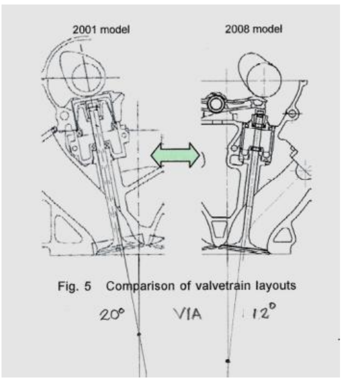
Design features

4 v/c; air-filled PVRS (with improved oil drainage); Finger cam followers (from 2002; previously inverted cup tappets) (see Fig. N121A).
VIA = *12*^{0} (was 20^0); Small compound (longitudinal) valve angle since 2003 (reduced combustion angle by 5^o).
Titanium Aluminide (TiAl) valves from 2002** (%ages: Ti 53.9, Al 42, Cr 2.5, Nb 1, Ta 0.5, B 0.1);
= Stem diameter 4.5 mm (reduced from 6.6 with previous Ti alloy).Together with the finger followers the reciprocating valve gear mass was reduced by 34% from 2000.
Linerless block (from 2003) (Al-alloy with Nikasil coating). Siamesed bores from 2004.
Piston since 2004 Metal Matrix Composite (MMC =Aluminium-alloy with 25% dispersed 3µm SiC particles)** (contributed to a reduction of piston mass from 255g to 210);
PH/B = 0.48; PH/S = 1.16. TiAl gudgeon pin** (stiffer with 17% part mass reduction).
Plain crank bearing SiCu liners with improved heat conduction properties.
Front end-oil-feed crank.
Crank pin (CP) dia. = 34 mm; Main journal (MJ) = 46 mm; Gudgeon pin (GP) = 17 mm
CP/S = 83.9%; CP/MJ = 73.9%; GP/CP = 50%.
Double fuel injectors per cylinder since 2004**.(see Fig. N121B for 2004 layout with 20^0 VIA).
It almost goes without saying that "Diamond-Like Carbon" (DLC) coating was applied to reduce friction wherever there was high rubbing-contact pressure.

See Fig. 121C on P.4.

**Banned by FIA for the 2006 formula.

Fig. N121A



It is assumed that the 2005 design was the same as 2008 shown and that the valve axes were symmetrical

Note the change in cam profiles

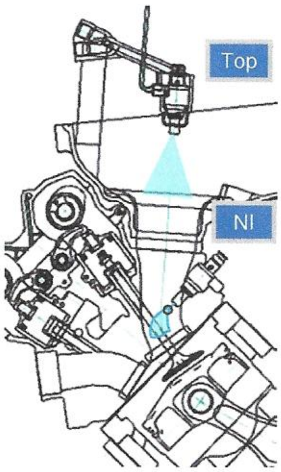


Fig. 15 Injector layout 2

Fig. N121B
NI = Near Inlet.

It is hoped that there will not be any objections to use of pictures here in a not-for-profit site whose intention is to aid study.

The 2014 Formula

The FIA formula for 2006 changed from the 11-year-old 3L capacity, with a maximum number of cylinders since 2001 of 10, to 2.4L with a mandatory 90^0V8 engine. Furthermore, the maximum Bore was to be 98 mm and the weight to be a minimum of 95 kg. The effect of the last restriction is shown by a Honda estimate that 78 kg (18% lighter) was technically possible. Various material bans were imposed, as shown above regarding the Honda 2005 specification. The requirement that engines must last for 2 events without overhaul, introduced for 2005 and assessed by Honda as needing 1500 km racing miles, was continued (grid penalties were applied otherwise).

In 2007 two further significant rules were introduced:-

Maximum(“Red line”) RPM of 19,000;

Major content of specification to be frozen at end February.

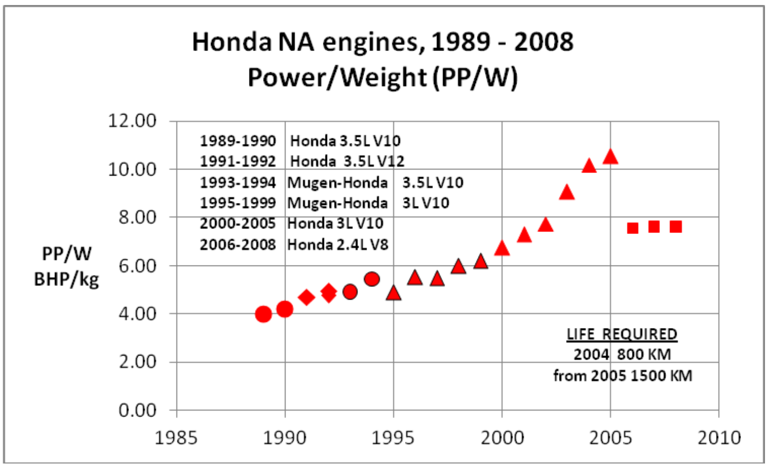
These 2006 – 2007 restrictions were novelties in Grand Prix rules (since 1908 regarding Piston Area, see The Sporting Limits).The steady trend of the FIA towards controlling **everything** was commented on by Honda in stating that the regulations (for chassis and engine) had doubled in letter content from 1990 to 2000 and **trebled over 1990 to 2008!**

Honda 2.4L V8 engines, 2006 – 2008

Year	2006	2007	2008
Type	RA806E	RA807E	RA808E
Confign.	90V8	90V8	90V8
B mm	97	97	97
S mm	40.52	40.52	40.52
B/S	2.394	2.394	2.394
V cc	2,395	2,395	2,395
Life Required – km	**1500**	**1350**	**1350**
PP – BHP	*718*	*724*	*724*
@ NP – RPM	*19,500*	*18,500*	*18,500*
BMPP – Bar	13.76	14.62	14.62
@ MPSP – m/s	26.34	24.99	24.99
W – kg	95	95	95
PP/W – BHP/kg	7.56	7.62	7.62

Advances in Power /Weight ratio over 1989 -2005

The chart below illustrates the truly remarkable advance In PP/W produced by Honda over 1989 to 2005, which was slowed by the FIA life rules in 2004/05 and halted in 2006.



Conclusions

Although far from powering a ”Grand Prix Car-of-the-Year” the Honda racing engines development story over 2000-2008 provides very many technical details of fairly-recent practice beyond the basic review of this website. It has been summarised in this Note 121 for that reason. This generous publication of data is in Honda’s tradition set with the RA168E of 1988 (DASO 20) and the RA122E/B of 1992 (DASO 69) which were used as sources for the website chapters quoted above.

It must be presumed that the successful CoY engines of the same period (Ferrari, Renault, Mercedes) displayed even more ingenuity than the remarkable efforts of Honda. These were only rewarded with one victory in 9 years (the 2006 Hungarian GP won by Jensen Button) even though BAR had the 1997 Champion as driver over 2000 -2003.

It will never be known if the double-diffuser car designed for 2009 under the technical direction of Ross Brawn would, with Honda’s own engine, have succeeded in that year if they had not retired in December 2008. As it was, with financial assistance from Honda, fitted with a Mercedes engine and re-branded as a “Brawn” it powered Button to the Drivers’ and the team to the Constructors’ Championships.

Fig. 121C
2005 Honda RA005E
90V10 B = 97 mm; S = 40.52 mm; B/S = 2.394; V = 2,994 cc
938 BHP @ 18,700 RPM

www.allf1.info/engines & Honda

P.S.

Further aborted technical advances

Honda made two technical advances to save mass in the engine “Bottom End” which are worth describing although the FIA ensured that one was short-lived and the other was not ready before it was banned:-

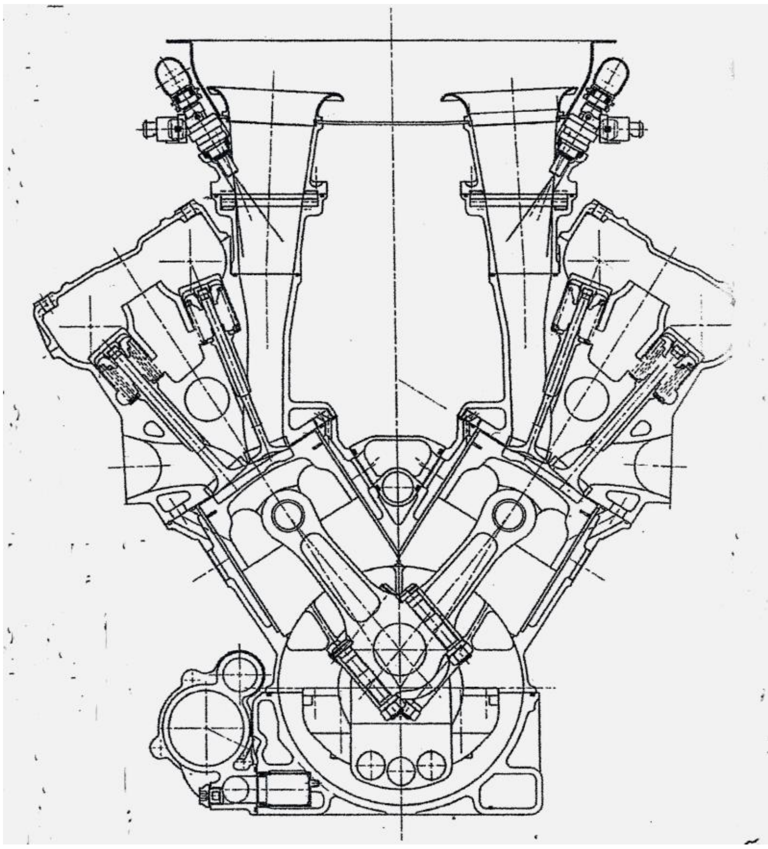
- Hollow connecting-rods.
  Honda had in 2000 used for con-rods a Ti-alloy (SP700) of 25% higher fatigue strength than the “workhorse” Ti6Al4V to reduce mass. To further improve the part, a hollow rod was then developed using 2 pieces diffusion-bonded. The process involved the machined pieces being heated to 70% of melting point then pressed together at 40 Bar in a vacuum for 5 hours. The joint disappeared. The assembly was then finish-machined.
  A much stiiffer rod with 8% lower mass than the previous I-section rod resulted, translateable to higher RPM. It was used in 2003 and 2004. The FIA then banned it.
- Hollow crankshaft
  Starting in 2004 a Honda programme produced a V10 crank in which the 34 mm pins had a 6 mm wall thickness. The joining process was Friction-welding. Five pieces were preheated and forced together in turn (rotating x stationary) at 1,900 RPM under a 10 tonne load. The joint strength was equal to the base material. The interior passage was cleaned to be suitable for oil-flow by barrel-finishing.
  The crank mass for the intended 2005 application, with added tungsten counterweights, was reduced by 7.8% from 10.3 kg for the solid assembly to 9.5.
  It was not ready for 2005 and the 2006 FIA rules banned multi-piece welded cranks.

SO30: Ferrari 1990 type 037

When the F1 formula became solely 3½L Naturally-Aspirated (NA) in 1989, Ferrari began to race a series of 65V12 engines with 5 valves per cylinder (5 v/c). This basic configuration, with increasing Bore/Stroke (B/S) ratio (see the 3NA Era Eg. 84) was retained by them for 4½ years until late 1993

The most successful engines of this series were the 1990 types 036 and 037, with which power in the type 641 car Alain Prost scored 5 wins and Nigel Mansell another, for 2nd place in the Constructors' Championship. Had it not been for the notorious collision between Prost and Senna (McLaren-Honda) at the 1st corner on the 1st lap of the 1990 Japanese GP it is *just* possible that the title could have been won. This result compares with 3 wins for the 65V12/5 v/c configuration in 1989 and *no further wins* after 1990. It was superseded by a Honda-influenced 65V12 4 v/c redesign. Clearly Ferrari had concluded that 3 inlet valves + 2 exhaust valves was not the "way-to-go".

A factory cross-section of a 1990 engine has recently been brought to the author's attention (see DASO 1124) and is shown on Fig. N122A below.



This drawing is dated 25/05/90 and is identified only as "F1-90". Ferrari introduced the type 037 for Qualification at the 7th race (French GP) in July 1990 (DASO 1123) so it seems reasonable to conclude that the section is of that type (sources often identify the 037 as a 1991 engine). This is therefore assumed in scaling for details (a process which involves some margin for error).

As one of only a few F1 section drawings published over the last 4 decades, although not the power of a "GP Car-of-the-Year" (CoY), it was decided to include it in the "Significant Other" series as SO 30.

65V12 B = 86 mm; S = 50.2 mm (DASO 1077)*

Swept Volume = 3,499 cc

B/S = 1.713

100/Smm = 1.992

*Source shows this data as 1991

Peak Power (PP) = 725 HP
@ NP = 14,500 RPM } (DASO 1077)*
Compression Ratio (R) = 13

so ASE = 0.642.

Inlet Valve Head Diameter (IVD) = *29.5 mm*

3 inlet valves, so $\left(\frac{IVA}{PA}\right) = 0.35$

Mean Gas Velocity at inlet @ NP (MGVP) = 69.3 m/s.

Coil Valve Return System (CVRS(with BNP = 20.8 m/s.

Valve Included Angle (VIA) = 20^0.

PP/V = 207.2 HP/Litre

BMPP = 12.79 Bar

@ MPSP = 24.26 m/s

ECOM = (EV x EC x EM) = $\left(\frac{BMPP}{38 \times ASE}\right)$ = 52.4%

Weight (W) has been reported as around 300 lb (136 kg) (www.gomog.com) but this is considered too light (the Honda 72V10 in the 1990 CoY is now known to have been 160 kg (it did have a balance shaft, weighing, say, 5 kg).

If W was actually, say, 150 kg, then PP/W = 4.8 HP/kg.

Con. Rod Length between centres (CRL) = *112 mm* (www.f1technical.net CRL for the 036 has been accepted); CRL/S = 2.23; Maximum Piston Deceleration @ NP (MPDP) = 7,221 g.

Piston Height (PH) = *52 mm;* PH/B = 0.6; PH/S = 1.04.

Crank Pin diameter (CP) = *34 mm*; CP/S = 0.68;
Gudgeon Pin diameter (GP) = *17 mm*; GP/CP =0.5.

Design features

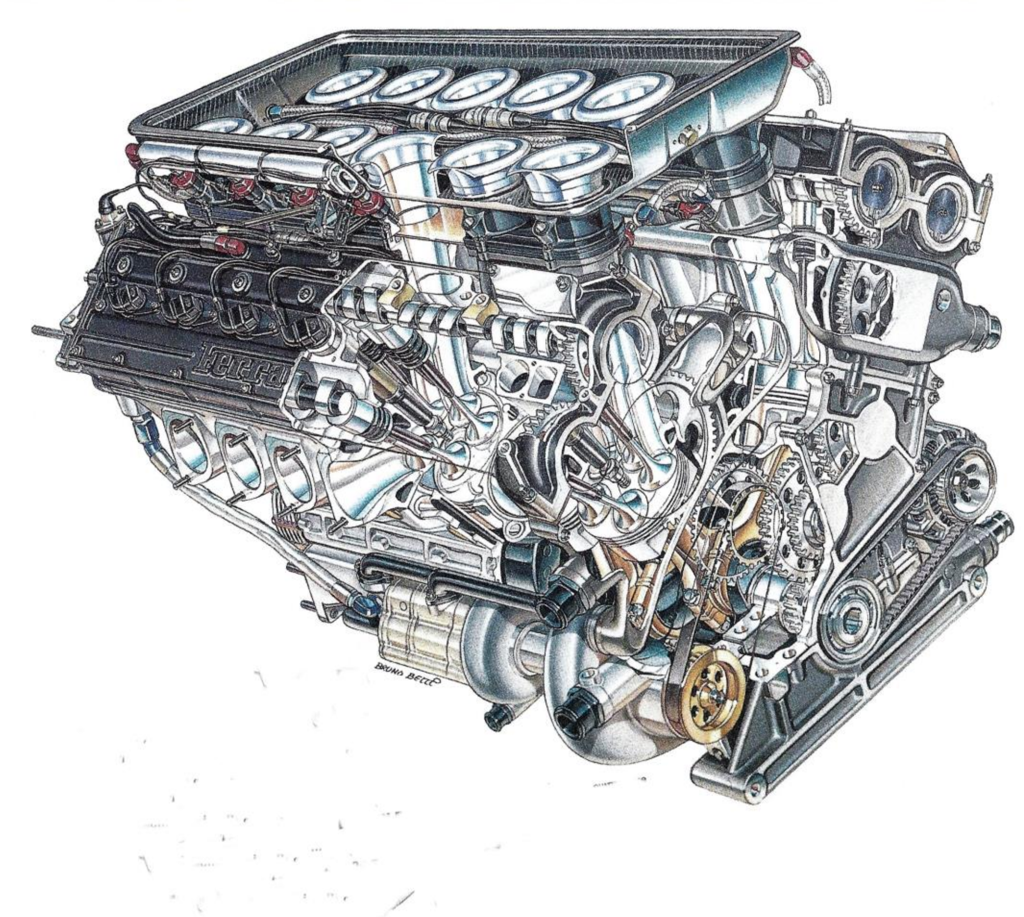
- Inlet tract length from entry to valve (LIN) was 222 mm; LIN/S = 4.42. Note 27 estimates the resonant MPS = about 20 m/s; this corresponds to 82% of MPSP, where PeakTorque would be expected. The area variation would affect this figure.
- The inlet tract was shaped to provide Tumble Swirl.
- The inlet tract had 60^0 of downdraft and the exhaust 30^0 of updraft, both relative to the plane of the piston crown.
- The combustion chamber was formed in the piston crown.
- Pistons had only 1 ring.
- Con-rods were I-section.
- Valve springs were half-shrouded by the tappets.
- Wet cylinder liners were fitted. Although this would usually imply that the block/crankcase was Al-alloy, it has been stated elsewhere that it was “ghisa molto speciale” (“special cast-iron”, probably spheroidal graphite material).

Fig. N122B (DASO 1125) on P.3 provides a cut-away of a later type of the Ferrari 65V12 5 v/c series.

Fig. N122B

Two observable differences from Fig. N122A are:-

- "Woods-type" tappets, mounted *above* the valve springs, giving them full exposure to cooling oil;
- 3 rings per piston.



References

DASO 1077 FERRARI; ALL THE CARS L Acerbi Haynes 2005.

DASO 1123 *AUTOCAR* GRAND PRIX REVIEW '90.

DASO 1124 Ferrari 1990 drawing published by Quattroruote in July 1990, copied to the author by courtesy of Ron Rex July 2015.

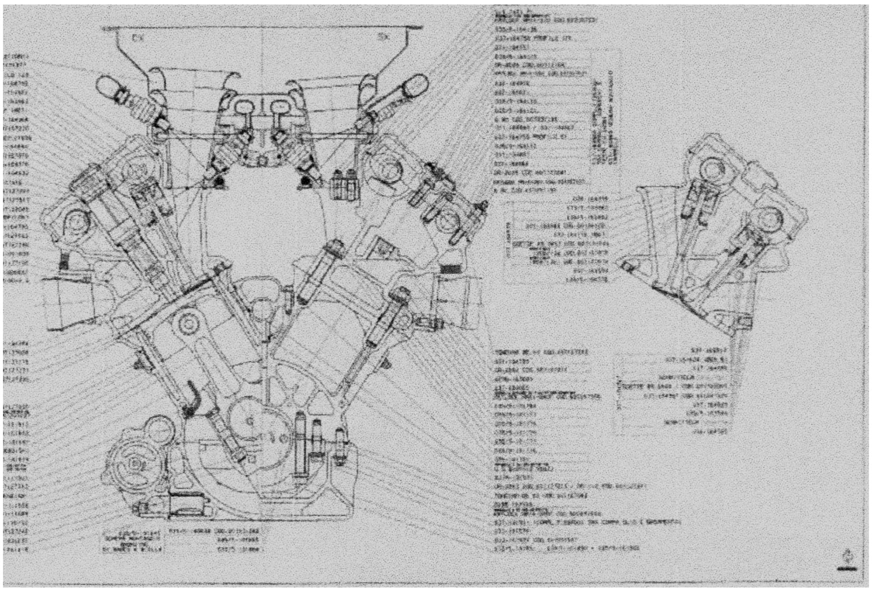
DASO 1125 Ferrari 65V12 cutaway published by Quattroruote, copied to the author by courtesy of Ron Rex July 2015.

Some correction & additions are now (August 2018) given as an extra page below.

Addition

DASO 1203 recently advised by correspondent Ron Rex (see refs. below) gives a cross-section of the 1990 Type 036 65V12 3.5L Ferrari engine, shown below. This was the engine used for the first half of the season.

Fig. N122C



[The source incorrectly refers to this as Type 037, although giving in its text the correct dimensions of the 036:-

84/52.6 mm = 1.597

The drawing has been confirmed as 036 by checking the B/S ratio.]

This section shows that the 3^{rd} central inlet valve was at 3^0 to the cylinder axis, where the other pair were at 16^0 . The pair of exhausts was at 6^0. It also shows the Variable Inlet System and twin fuel injection nozzles. Neither feature is shown on Fig. N122A, although it was a later engine. The drawing was presumably specially done for the magazine article in May 1990. It may be that Ferrari did not wish to disclose them at that time (Fig. N122C was published in 1993).

Correction

A further detailed reference (DASO 1198) showed that the move from 036 to 037 was different from that given above, with different powers, as follows:-

Races	**Type**	**RACE TUNE** **B x S mm Vcc** **BHP @ RPM**	**QUALIFICATION** **B x S mm V cc** **BHP @ RPM**
• First 9 USA to Germany	036	84 x 52.6 3,498 656 @ 13,200	
• Introduced for Q at 3^{rd} race (S. Marino)	037		86 x 50.2 3,499 671 @ 13,600
• Last 7 Hungary to Australia	037	681 @ 13,600	
• Introduced for Q at 10^{th} race (Hungary)	037		700 @ 13,800

Appendix 1 has been updated with the latest 037 data. ECOM = 53.7% instead of 52.4%.

References

DASO 1203 Designed for Speed Museum of Modern Art, New York 1993
Advised by courtesy of Ron Rex July 2018.

DASO 1198 Ferrari Monoposto, Catalogue Raisonné, 1948 – 1997 B. Alfieri Automobilia.
Advised by courtesy of Ron Rex.

<u>Note 123</u>
<u>Engine Weight as Installed</u>

This website has given the best available data on engine weight but there are reservations about its value. There is usually no definition of what is included, so that there is plenty of scope for suppliers to state artificially low figures. An amusing anecdote about this concerns the delivery in 1966 of a Flat-H16 3 Litre BRM engine to Lotus for their type 43 – two mechanics went out to the lorry bringing it from Bourne but they had to call for another two strong chaps to help unload it!

What *can* be said with certainty is that the figure published, which can be called the “Bare Weight” (WB) is nothing like the figure needed by a car designer. That is the “Installed Weight” (WI) ready –to-race, including everything necessary to operate it and keep it running for the required duration.

Very little is available in the public domain on the ratio WI/WB. This Note can give only 2 such figures, for widely-different applications, but it was thought that visitors would like to see them discussed.

<u>2013 Grand Prix engine WI/WB</u>

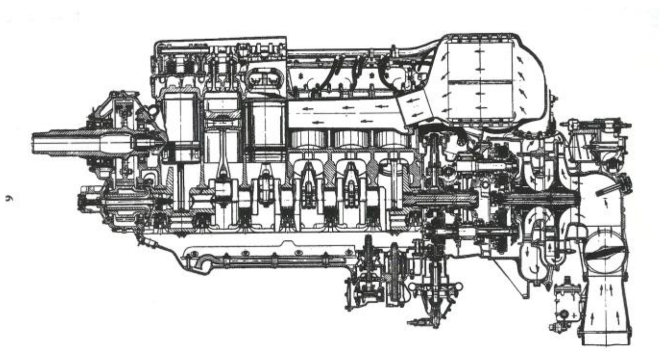
Renault RS27
The Championship 2013 Grand Prix engine

allf1.info

DASO 1134 (see refs. below) stated that *“A current* [2013] *V8 weighs 120 kg with all ancillary components (such as radiators) included*”. This was for a 90^0V8 2.4 Litre Naturally-Aspirated engine whose WB was by FIA rule a minimum of 95kg. Lighter engines could have been built if it had not been for that rule:- Honda estimated that they *could* have produced a 78 kg unit (DASO 1121). The genuine ratio of WI/WB would then have been:-

$$WI = 78 + (120 – 95) = 78 + 25 = 103 \text{ kg};$$
$$WI/WB = 103/78 = \underline{\mathbf{1.33.}}$$

<u>1943 Rolls-Royce (Packard-built) *Merlin* 100 Series WI/WB</u>



Rolls-Royce Heritage Trust

Rolls-Royce Merlin 100 Series Aero Engine

DASO 328 p.677 gives very complete data on the engine WB and the additions to WI (contd. P.2).

			Lb
(A) Type V1650-3 *Merlin* 60^{0}V12 27 Litre engine,			
2-stage mechanically-supercharged, intercoole d		1,639	
Deduct crank-speed reduction gear			
(on limited data),	say,	(100)	
	WB		1,539
(B) Engine Mounting & Cowl			284
(C) Cooling system }			
(D) Water & Oil }			652
(E) Cooling system mounting, ducts & controls			116
(F) Intake & Exhaust systems, Starter, Engine Controls,			
Propeller & its Controls		624	
Deduct Propeller, etc (from limited data)	say,	(324)	
			300
		WI	2,891

WI/WB = **1.88**

Discussion

Looking at the weight *items* for the aero engine versus the Grand Prix unit suggests the following as additions to the ancillary 25 kg quoted above-

(B). The GP engine also does duty as an integral part of the chassis, and the cowl is part of the body so neither item needs to be included.	
(C). Already included in the 25 kg.	
(D). Not included: add water at, say, 10 Litres @ 1 kg/L	10 kg
Add oil at, say, 3 Litres usage (100km/litre (1121)) + 200% = 9 litres total @ 0.9 kg/L, take	8 kg
(E). Included in the 25 kg.	
(F). The GP engine has a substantial carbon fibre intake duct: add, say,	2 kg
The tuned exhaust systems are substantial pieces of pipework, add (from DASO 1121)	15 kg
The starter should be included in the 78 kg.	
Additional weight	35 kg

Then WI/WB = (78 + 25 + 35)/78 = 138/78 = **1.77**

The author will be pleased to have factual comments on this deduction.

References

DASO 328. The Development of Aircraft Engines and Fuels. R Schlaifer & S Heron. Boston. 1950.
DASO 1121 Honda R&D website Technical Review F1 Special. December 2009.
Source advised to DST by courtesy of Ron Rex, July 2015.
DASO 1134. *F1 Racing,* September 2013.

Note 124

Performance of different types of Naturally-Aspirated (NA) 4-stroke 4-cylinder petrol engines Series Production to Full Racing, 1953 - 1979

Performance comparisons for NA 4-stroke 4-cylinder petrol engines ranging from Series Production to Full Racing were published by the Ford engineer C. Brewer in the *Automotive Technology Series* Vol. 2 1968 in a contribution on *Engine Design* (DASO 939). A chart of Brake Mean Effective Pressure (BMEP, psi) v. Crank Speed (N, RPM) was provided for examples from the plainest Side-Valve (SV) Series Production engine, through various Overhead Valve (OHV) Production types [Push-Rod (PROHV), Single Overhead Camshaft (SOHC) and Double Overhead Camshaft (DOHC)], to Full Racing SOHC and DOHC engines. Naturally, most data were for Ford engines or engines with re-designed cylinder heads on Ford bottom ends, but one Coventry Climax engine was included because it had a wedge combustion chamber.

This Note 124:-

(a) reworks the data in the standard format of this website as BMEP (Bar) v. Mean Piston Speed (MPS, m/s) and adds Power (P, HP) v. Crank Speed (N, RPM);

(b) extends the 7 examples given by Brewer with 3 more Racing engines.

The source provided minimum details of each engine and so they have had to be researched for this Note.

The highest value of BMEP reached in this sample, by the BMW M12/7, was 15.4 Bar @ 21.3 m/s in 1979, which has only been exceeded fairly recently.

Engine Examples

The salient features and performance data of the 10 engine examples are given below. The performance curves are shown on P.4.

Tabled data

All NA 4-stroke IL4 petrol* engines

Swept Volume (V) cc	Peak Power (PP) HP @ Crank Speed (NP) RPM	PP/V HP/Litre	BMPP** @ MPSP / BMTP** @ MPST Bar m/s
1. **1953 Ford 100E**	Production SV with Ricardo-type squish combustion chamber. Cast iron head. 1 carburetter. Same side inlet and exhaust ports. Siamesed inlets. 240^0 inlet cam period (IOD).- 3-bearing crank. A section of the 100E is shown on P.5		
2.5''/3 41/64'' = 0.687 (63.5 mm/92.472)			
1,172	36.2 @ 4,500	30.9	6.14 @ 13.88 / 7.72 @ 7.71

The Sales brochure claimed 30 HP, so this must have been a “blue-printed” engine.

Continued on P.2

*Varying grades of petrol. Say, 80 Octane for Production and about 100 for Racing.

**BMPP & MPSP = BMEP & MPS @ Peak Power

BMTP & MPST = ” ” ” Peak Torque

Swept Volume (V) cc	Peak Power (PP) HP @ Crank Speed (N) RPM	PP/V HP/Litre	BMPP @ MPSP BMTP @ MPST Bar m/s

2 **1958 Coventry Climax FWE** Low Production. SOHC with in-line valves inclined in "Wedge" combustion chamber. Al-alloy head (and block). Same side inlet and exhaust ports. Stage III uprating for Lotus L14 *Elite* sports coupe with 2 x 2-choke Weber carbs. IOD =-248^0 . 3-bearing crank.

A section of the FWA, from which the FWE was derived, is given on P.5.

3"/2 5/8" = 1.143
(76.2 mm/66.675)

1,216	89.7 @ 6,000	73.8	11.00 @ 13.34 12.07 @ 10.00

1962 Ford 115E Production PROHV for C*ortina* with vertical in-line valves in a "Bath-tub" combustion chamber. Cast iron head. 1 carb. Same-side inlet and exhaust ports. IOD = 248^0 . -5-bearing crank (1st engine family member, 105E, was 3-bearing).

A section of the 105E, from which the 115E was derived, is shown in Note 81.

3 3/16''/2.29'' = 1.392
(80.9625 mm/58.166)

1,198	50.3 @ 5,000	42	7.52 @ 9.69 9.41 @ 5.23

4. **1962 Cosworth-Ford FJ** Basically a 1959 Ford 105E Production engine (No. 3 in same family. and similar in design). Developed for Formula Junior racing, with IOD = 320^0 and 2 x 2-choke Weber carbs. Bore enlarged in 1962 from 3 3/16'' (80.9625 mm) to 85 mm for alternative weight class. 3-bearing crank.

See also Note 81 for 105E section.

85 mm/1 29/32'' = 1.755
(48.419 mm)

1,099	98.4 @ 7,500	89.5	10.68 @ 18.10 11.50 @ 8.88

5. **1962 Lotus-Ford "L26"** Low Production Al-alloy DOHC re-designed head on a Ford 116E bottom end.- Valves at 54^0 included angle (VIA). 2 x 2-choke Weber carbs. Tuned for Lotus L26 *Elan* sports car, with IOD = 248^0.

A section of the engine is shown in "ILLUSTRATIONS for Appendix 6" at P.17.

3 3/16''/72.75 mm = 1.113
(80.9625 mm)

1,498	91.1 @ 5,500	60.8	9.39 '@ 13.34 11.58 @ 9.94

6. **1964 Cosworth SCA** Al-alloy SOHC re-designed head on a Ford 116E bottom-end but with 105E-size special crank throw for Formula 2 racing. Vertical in-line valves in flat head with combustion chamber in the piston. Same side inlet and exhaust ports. 50^{0}downdraft inlet ports. 2 x 2-choke Weber carbs. IOD = 324^0 . 5-bearing crank.

A section of the cylinder head is given in Note 73.

.3 3/16''/1 29/32'' = 1.872
(80.9625/48.419)

997	122.7 @ 8,500	123.1	12.96 @ 13.72 13.69 @ 12.27

Continued on P.3

Swept Volume (V) cc	Peak Power (PP) HP @ Crank Speed (N) RPM	PP/V HP/Litre	BMPP @ MPSP / BMTP @ MPST Bar m/s

7. **1965? Lotus-Ford "L23"** As no. 5 but fully-tuned for racing, probably by Cosworth. No details are known for the engine whose BMEP was given by Brewer, but it would have had larger valves (possibly +6% area – Wikipedia), higher lift and a longer IOD cam (possibly as much as 320^0).
The power curve carries on from the mildly-tuned engine.
The designation as "L23" is arbitrary, assuming it would have been raced in that Lotus.
3 3/16''/72.75 mm= 1.113
(80.9625 mm)

1,498	146.6 @ 7,300	97.9	12.00 @ 17.70 13.00 @ 15.28

8. **1966 Cosworth FVA** Al-alloy DOHC 4 valves per cylinder re-designed head on a Ford 120E bottom-end for a new Formula 2. VIA 40^0. Inlet ports shaped by Keith Duckworth to give "Barrel Turbulence" (also known later as "Tumble Swirl", although the concept was kept secret for many years). IOD = 320^0 . Port fuel injection. 5-bearing crank.
(See "The Unique Cosworth Story" and Note 26).
3.3/8''/2.722'' = 1.24
(85.725 mm/69.139)

1,596	222 @ 9,000	139.1	13.83 @ 20.74 14.50 @ 16.13

9. **1967 Ford "Cross-Flow"** Production PROHV for C*ortina* with vertical in-line valves in a flat head with the combustion chamber mostly in a bowl in the piston top. Cast iron head. Exhaust ports on opposite side to the inlets. 1 carb. IOD = 248^0 . 5-bearing crank.
A section of the 1,600 cc version of this engine is shown on P. 5.
3 3/16''/72.75 mm = 1.113
(80.9625 mm)

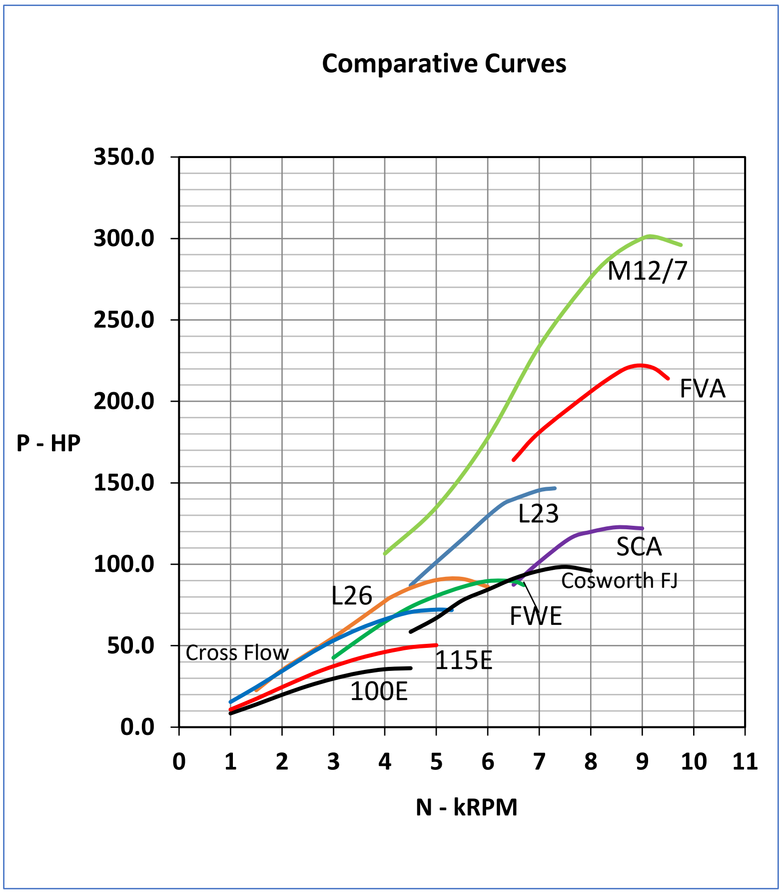
1,498	72.1 @ 5,000	48.1	8.62 @ 12.12 10.62 @ 6.55

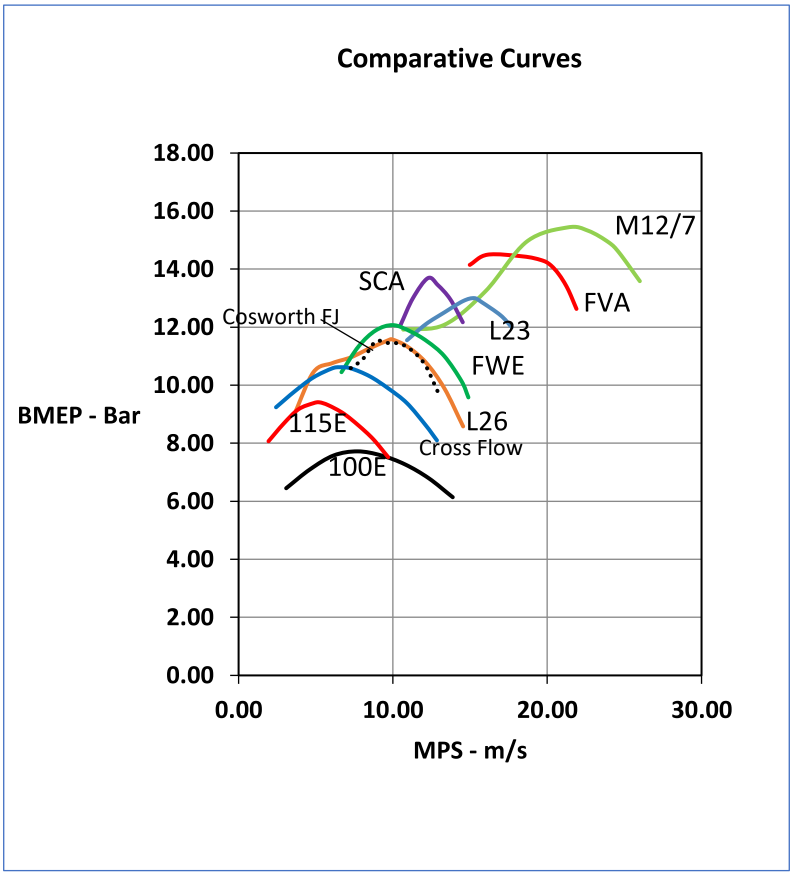
10. **1979 BMW M12/7** Al-alloy DOHC 4 valves per cylinder re-designed head on a Production cast iron block bottom-end for a new Formula 2. VIA 40^0. IOD = probably 320^0 . Port fuel injection.
See "Significant Other" at Fig. SO19A for an illustration of this engine.
Influenced by Duckworth architecture and replaced the unsatisfactory Apfelbeck radial-valve design (see "How many valves per cylinder" at p.11).
89.2 mm/80 = 1.115

1,999.7	301 @ 9,250	150.5	14.56 @ 24.67 15.44 @ 21.33

Performance curves for engines nos. 1 to 10 are shown on P. 4.

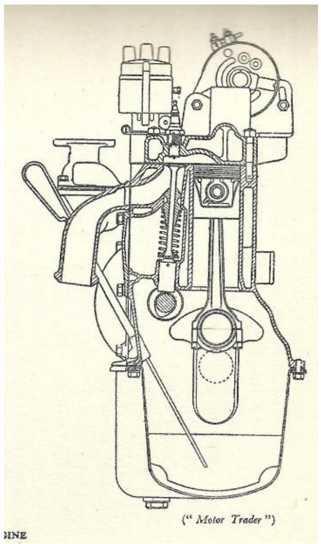
Analysis of the Flexibility (FY) of the engines is continued on P. 6.

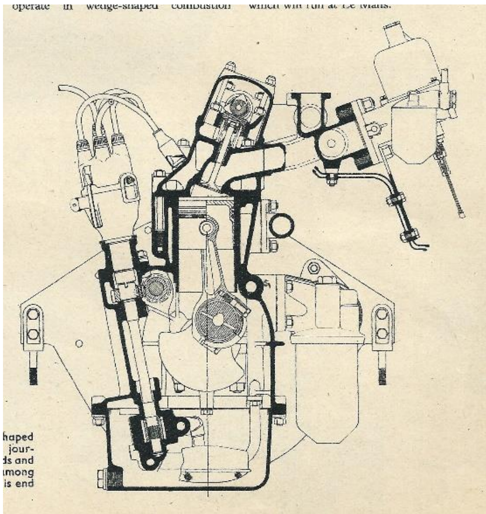

Comparative Curves
350.0
300.0
250.0
200.0
150.0
100.0
50.0
0.0
P - HP
0
1
2
3
4
5
6
7
8
9
10
11
N - kRPM
M12/7
FVA
L23
SCA
Cosworth FJ
FWE
L26
Cross Flow
115E
100E


Comparative Curves
18.00
16.00
14.00
12.00
10.00
8.00
6.00
4.00
2.00
0.00
BMEP - Bar
0.00
10.00
20.00
30.00
MPS - m/s
M12/7
SCA
Cosworth FJ
FVA
L23
FWE
115E
L26
Cross Flow
100E

Ford 100E

DASO 337

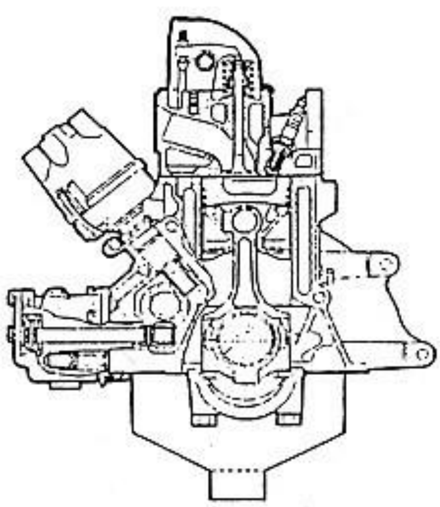




Coventry Climax
FWA
This engine, B/S = 2.85’’/2 5/8’’ = 1.086
72.39 mm/66.675 1,098 cc,
was the original automotive application.
The FWE was derived from it with enlarged bore:-
B/S = 3’’/2 5/8’’ = 1.143,
76.2 mm/66.675 1,216 cc.
DASO 135

Ford Crossflow
This is the 1971 version ``
enlarged to
B/S =3.188’’/3.056’’ = 1.043
80.975 mm/77.622 1,599 cc.

thekentlives.com



Flexibility

Flexibility (FY) is defined here for piston engines as the drop of RPM between Peak Power speed and Peak Torque speed as a ratio of the former. Expressed in Mean Piston Speed, as tabled:-

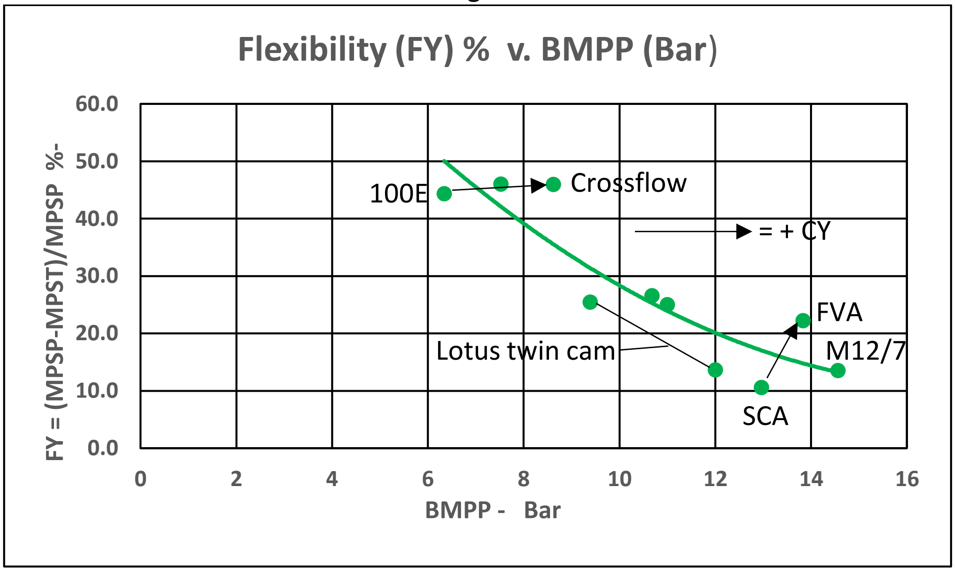
$$FY\ \% = \frac{MPSP - MPST}{MPSP} \times 100$$

The author has investigated a "Natural Rule of Thumb" (which can be accessed) expressed by:-

$$\frac{\text{Effectivity (EY)} \times \text{Flexibility (FY)}}{\text{Complexity (CY)}} = \text{Constant.}$$

The 10 engine data sample, taking EY = BMPP, enables this relation to be examined. The result is shown on Fig. 1

Fig. 1

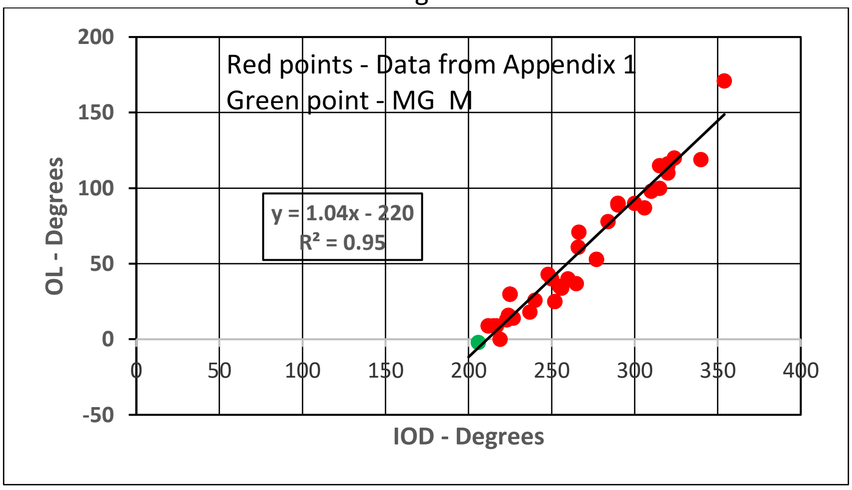


Since the data is the result of designers with different methods, there is scatter but the general relation does follow the "Rule". In particular the Lotus Twin Cam at two different states of tune shows it well. Some scatter is actually the result of different CY; egs., the Ford SV 100E relative to the Ford Crossflow; also the Cosworth SCA versus the Cosworth FVA. These increases in CY enabled higher BMPP to be obtained at the same or higher FY.

Other ways in which increased CY can raise the (BMPP x FY) product are:- Variable Valve Timing; and Variable Inlet Length; but these were not available to the sample engines.

A major cause of decreased FY as BMPP is raised is the increasing inlet and exhaust valve overlap (OL). This is an inevitable but not determinate result of increasing Inlet Open Duration (IOD), as shown on Fig. 2.

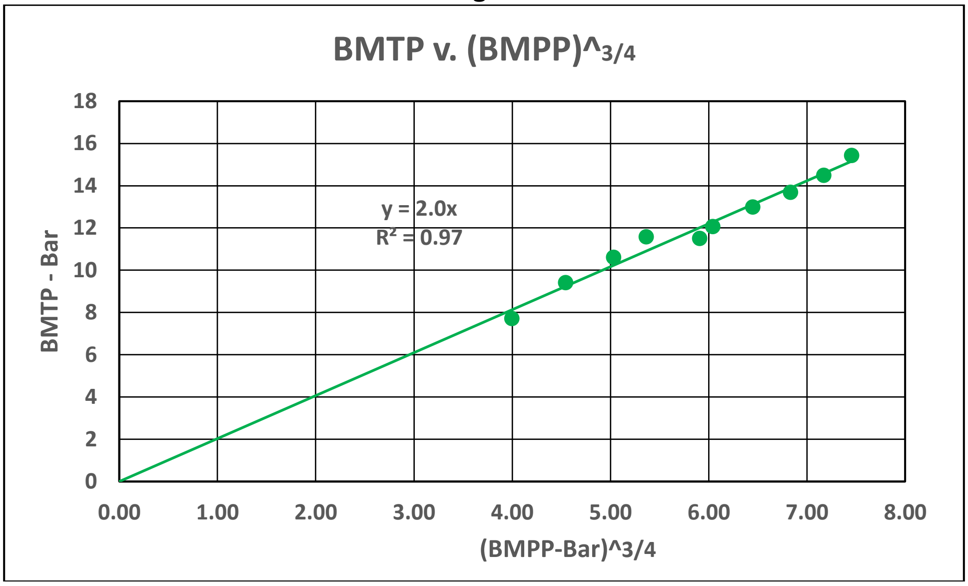
Fig. 2



<u>Relation between BMPP and BMTP</u>

The relation between BMPP and BMTP was also examined with the 10 engine sample. The result is shown on Fig.3.

Fig. 3



Derek S. Taulbut

February 2017.

Note 125. Historical Mean Piston Speed (MPS) for reciprocating steam locomotives

Mean Piston Speed (MPS) has been much used in this review for the reason explained in Note 13 Part 1. It was thought that a historical look at this factor would be of interest to viewers.

It was recognised quite early in the development of reciprocating steam locomotive engines that MPS was important in the stressing of the piston and connecting rod. With these parts always in ferrous material, and so of the same density, their stress is proportional to (MPS)^2.

In standard steam engine design the piston, being double-acting, is supported by a cross-head outside the cylinder. Therefore it is not subjected to side loads and can be very short and just sufficient to carry sealing rings. In the 1907 GWR " *Star* " class (for which a drawing is available, thanks to GWR and " *Childrens' Encyclopedia* "!) the axial length was only 11% of the stroke.

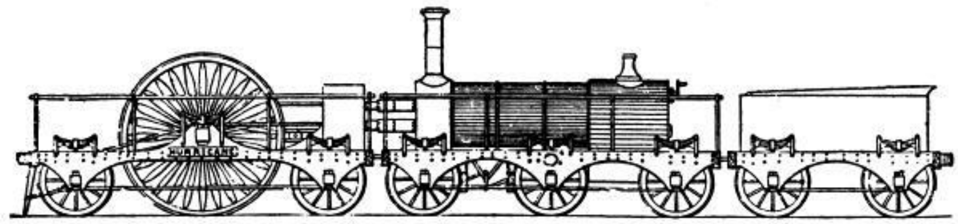
For locomotive steam engines travelling at V MPH on driving wheels of D feet in diameter and with a piston stroke of S feet the relation is

$$MPS = 56.02 \times (S/D) \times V \quad \text{ft/min.}$$

Examples

(1). In 1838 Isambard Brunel ordered a pair of locomotives from an outside builder for the GWR and specified that MPS should not exceed 280 ft/min (1.4 m/s) at 30 MPH (Ref. 1). This for cast iron pistons at something under 140C and with wrought iron connecting rods. Actually, Brunel's specification could only be met by having the cylinders carried on their own set of wheels, with the crank geared-up 2.3:1 to the driving pair, and with the boiler carried separately on another-truck (see Fig. 1 below). The result was described as " *More of a procession than an engine!* ". The locomotives were not a success. It must have been difficult to make steam-tight flexible joints in the connecting pipes. The adhesive weight was minimal. Judging by later values of MPS, Brunel seems to have been excessively cautious, rather surprising for such an innovator.

Fig. 1.

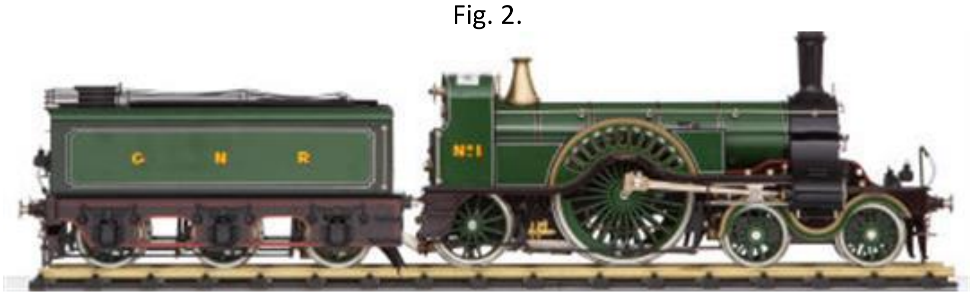


W FIG. 6. HARRISON'S "HURRICANE."

wikipedia

(2). To keep stresses within the available material limits at the design and manufacturing techniques of each age, as required speeds rose, the driving wheel diameters were increased. In 1870 the GNR Chief Engineer, Patrick Stirling, used 8 feet 1 inch single wheels to keep MPS to about 1,200 ft/min (6.2 m/s) at express speeds of 75 MPH (Ref. 1).

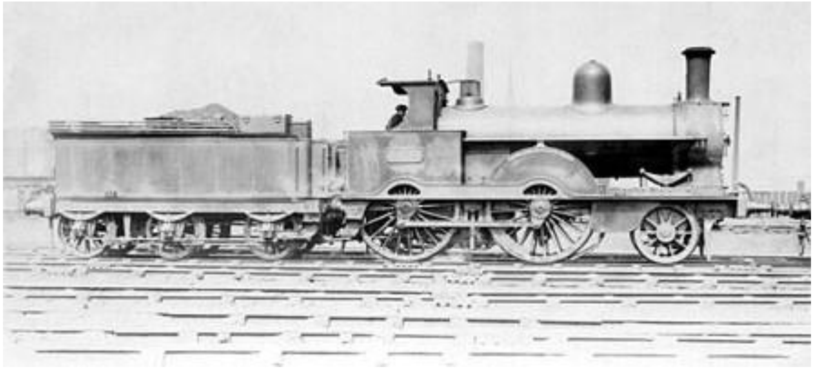
Fig. 2.



thesaleroom

(3). By 1875, during the last " Race to the North ", the LNWR locomotive " Hardwicke ", designed by Francis Webb, reached 88 MPH at about 1,400 ft/min (7.1 m/s) (Ref.1).

Fig. 3.



Wikipedia

(4). In 1904 the GWR locomotive " *City of Truro* ", designed by William Dean with George Churchward's assistance, was timed unofficially (but by an experienced observer) to reach 100 MPH (<u>down a 1/90 slope</u>) (Ref. Wikipedia). This was 1,809 ft/min (9.2 m/s).

Fig. 4.

modelrailforum

(5). An officially timed dynamometer run in 1934 by the LNER " *Flying Scotsman* ", designed by Nigel Gresley, recorded 100 MPH (probably down Stoke Bank, 1/200). This was 1821 ft/min (9.2 m/s). The engine had Ni,Cr-alloy con. rods (Ref. 3).

Fig. 5.

Wikipedia

This shows the corridor connection on the tender, which enabled crews to change on non-stop London to Edinburgh runs.

(6). In 1935, about a century after Brunels' specification, AlCo built 4-4-2 locomotives to haul the " *Hiawatha* " express between Chicago and Minneapolis on the Chicago, Milwaukee, St Paul and Pacific Railroad at a regular 100 MPH (Ref.2). This was an MPS of 1,900 ft/min (9.6 m/s). These engines had 300 psi steam pressure, instead of the then-usual 275, especially to keep down the diameter and mass of the forged-steel pistons, working at a superheated steam temperature of 370C. The con. rods were Ni-alloy steel.

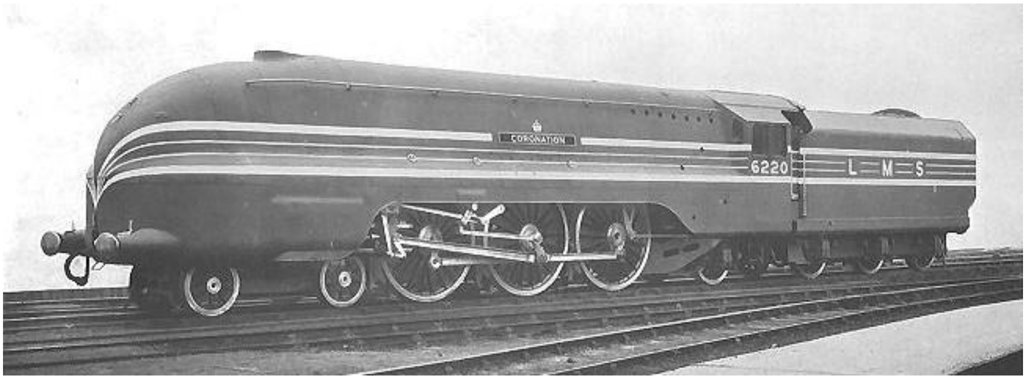
See Fig. 6 on P.3

Fig. 6.

Pinterest

(7). When the LMS 4-6-2 engine “ *Coronation* ”, designed by William Stanier, reached 114 MPH in June 1937 (down a 1/269 slope) the level of MPS was 2,200 ft/min (11.2 m/s) (Ref.3). The superheated steam temperature was 312C.

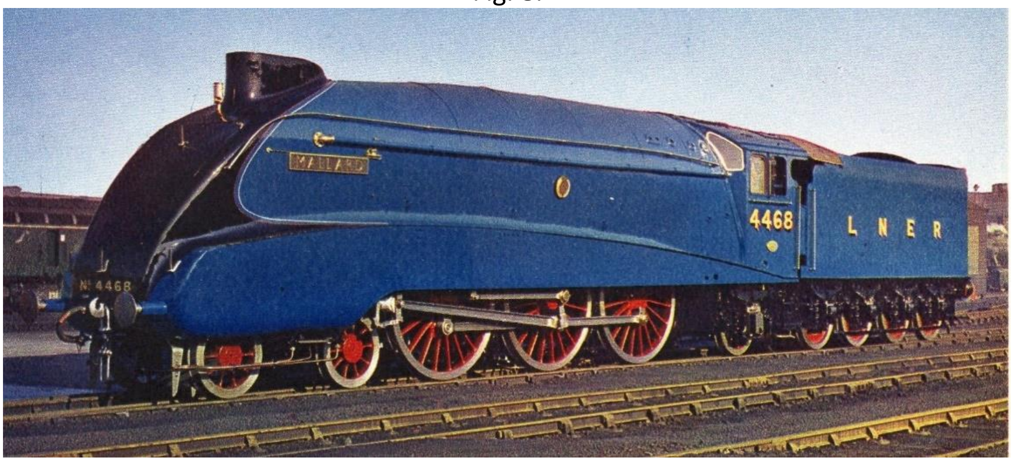
Fig. 7.



railalbum.co.uk

(8). In July 1938 the LNER A4-class 4-6-2 “ *Mallard* ”, designed by Nigel Gresley, set an all-time steam record of 126 MPH (down the 1/200 Stoke bank). MPS was 2,300 ft/min(11.6 m/s). This was at the cost of a failed middle big-end bearing (a well-known A4 fault caused by wear in the gear operating the central cylinder valves from the outside cylinders, so that this cylinder was over-loaded). (Ref. 3).

Fig. 8.



pinterest

(9). Post-WW2 a Norfolk & Western J-class 4-8-4 locomotive, running on a " race-track " section of the Pennsylvania, achieved 110 MPH. This was an MPS of 2,818 ft/min (14.3 m/s). The stress would have been 50% higher than " *Mallard* ". The con. rods were " lightweight ". All bearings were roller. The boiler was mechanically stoked. (Ref. 4).

Fig. 9.

accucraftukltd

(10). As a tailpiece, not part of the rising MPS series above, a BR class 9F 2-10-0, designed by R Riddles for heavy freight service, was timed unofficially at 90 MPH while descending the 1/200 Stoke Bank in 1958 on a passenger train (Ref. worldrailfans.info). MPS = 2,353 ft/min (12 m/s), i.e ,slightly more than " *Mallard* ". The bearings were plain and, to avoid excessive wear to these, such high speeds were banned later..

Fig. 10.

traintalktv

This engine was not named while in BR service.

<u>Conclusion</u>

The increase in reciprocating parts stresses, proportional to (MPS)^2, over the period 1870 to 1904 was [9.2 m/s/6.2 m/s]^2 = 2.2. From 1904 to 1950 it was [14.3/9.2]^2 = 2.4. This was with locomotives designed for normal use, although no doubt carefully prepared, not " racing " engines. The changes in materials were, egs. pistons improving from cast iron to forged steel, and con. rods from wrought iron to plain steel to Ni,Cr-alloy steel.

<u>Sources</u>

(1). The Illustrated Encyclopaedia of the World's Steam Passenger Locomotives.
B. Hollingsworth. Salamander. 1982
(2). Loco Profile No. 26. The Hiawathas. B. Reed. Profile Publications. 1972.
(3). British Pacific Locomotives. C. Allen. Ian Allan. 1971.
(4). Loco Profile No. 20. The American 4-8-4. B. Reed. Profile Publications. 1972.

Note 126. Origins of Crank, Cam, Con.-Rod, Poppet-Valve and Steel Wire Coil Spring

Brief notes on the origins of these essential components of virtually all automobile Grand Prix engines (until pneumatic valve springs were in vented by Jean-Pierre Boudy in 1984 (474)) may be of interest. The source is (630: see Ref. below) unless given otherwise.

The Chinese knew of the **CRANK** in 100 BC, eg. for turning a hand mill.

The **CAM** to produce intermittent uni-directional motion from rotary motion appeared in the 10^{th} C, with a water-wheel-driven camshaft lifting a trip-hammer to permit forging operations under the gravitational return motion.

The 1^{st} **CRANK-PLUS-CONNECTING-ROD** mechanism, to produce reciprocating motion from rotary appeared in the 15^{th} C, a water-wheel working a saw or blacksmith's bellows. The **inversion** of this principle to convert reciprocating motion into rotary was Patented by James Pickard in 1780. He applied it to a Newcomen-type steam engine.

The (weight-loaded) **POPPET-VALVE** was invented by Denis Papin as a safety device on his pressure-cooker in1679.

The **COIL SPRING** was first Patented in 1763 by R. Tredwell (Wikipedia), and was originally used in upholstered furniture. Later it was used to load a poppet valve to provide a safety system for boilers. It needed **STEEL WIRE** to be effective and this became available after it was Patented in America in 1857 (Wikipedia).

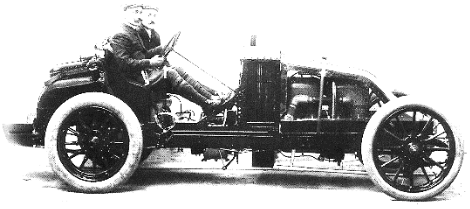
Reference

DASO 630. The Inventions that changed the World. Consultant editor G. Taylor. Readers Digest 1982.

Note 127 A selection of Grand Prix "Cars-of-the Year" (CoY)

Full tabulated performance details and analyses for each engine are given in Appendix 1.
Captions refer to those engines which are in the website list of CoY.

- **1906 Renault. The 1st Grand Prix engine.**
  French Grand Prix winner: Ferenc Szisz



motorsporthistory.ru

See 1st Naturally-Aspirated Era (1NA) at PP 1 to 2 and Addendum after P.18.

- **1924 Alfa Romeo. The 1st Mechanically-Supercharged engine.**
  French Grand Prix winner: Giuseppe Campari

pinterest

See 1st Pressure-Charged Era (1PC) Part 1 at PP 1 to 3
And Corrections & Additions at P.12.

- **1939 Mercedes-Benz. The 1st 2-stage Mechanically-Supercharged engine.**
  1939 European Champion Hermann Lang
  (driver shown is Manfred von Brauchitsch)

daimlermedia

See 1st Pressure-Charged Era (1PC) Part 1 at PP 31 to 38

Continued on P.2

- **<u>1951</u> Alfa Romeo. The last Mechanically-Supercharged engine.**

1951 Champion Juan Fangio

365daysofmotoring

See 1st Pressure-Charged Era (1PC) Part 2 at PP 1 to 5.
And Corrections & Additions at PP 13 to 14.

- **<u>1955</u> Mercedes-Benz. The most expensive engine (in real money terms) before 2014.**

1955 Champion Juan Fangio

daimlermedia

See 2nd Naturally-Aspirated Era (2NA) Part 1 at PP 4 to 8.

- **<u>1960</u> Cooper-Climax The Cooper mid-engined revolution.**

1960 Champion Jack Brabham

simonlewis

See 2nd Naturally-Aspirated Era (2NA) Part 2 at PP 6 to 12.
And Note 66 Standard GP, Note 66 Illustrations, and Note 66B

Continued on P.3

- **<u>1978</u> Lotus The ground-effect revolution.**

1978 Champion Mario Andretti

italianguide

<u>See</u> Eg. 47 The Unique Cosworth Story at P.18.
<u>And</u> Corrections & Additions at PP 18, 30, and 40,41.

- **<u>1988</u> McLaren-Honda. Last TurboCharged engine until 2014.**

1988 Champion Ayrton Senna

pinterest

<u>See</u> 2nd Pressure-Charged Era (2PC) Egs. 69, 70, 71 at PP 6 to 9 with power curves and fig.70A and figs. 71A, 71B.
<u>And</u> Corrections & Additions at PP. 56 to 57.

- **<u>2000</u> Ferrari. Their 1st Champion-driver mount in 21 years.**

2000 Champion Michael Schumacher

wikimedia

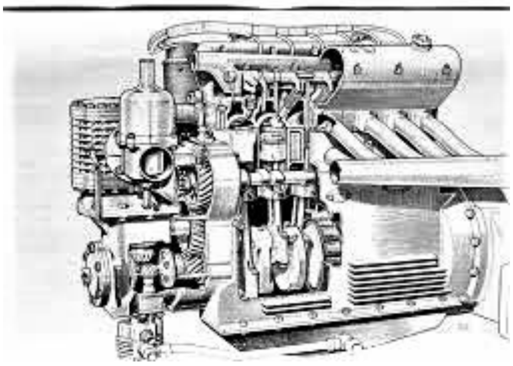
<u>See</u> 3rd Naturally-Aspirated Era (3NA) Part2 Eg.84 and Eg. 85 at Eg. 85 PP 1 to 7.

Note 128

The ERA crankshaft centre bearing

In 1933 Murray Jamieson designed a new stronger crankshaft for the 1.5 litre 6-cylinder "White Riley", conceived by Raymond Mays and sponsored by Victor Riley (DASO 446). This was to be Pressure-Charged by a Roots-type blower, also of his design. He had to adopt the 3-bearing layout of the "donor" Riley engine crankcase and his solution for the centre bearing was -and still is – unique.

A pair of full-diameter (6''?) circular webs supported a small number (12?) of large diameter (1 ¼''?) rollers, made by the American firm of Hyatt. These *may* have been built-up to have some radial resilience. The shaft with this assembly was inserted end-on into an outer race mounted in the crankcase. This description can best be followed in the figure below.



riley-prewar-specials.com

The new crankshaft basic design was incorporated in 1934 into the first, and all subsequent, ERAs. It therefore coped with strokes varying from 2 ¾'' (69.85 mm, for 1.1 litres) through the "standard" 3 ¾'' (95.25 mm, for 1.5 litres), to 4 13/64'' (107.76 mm, for 2 litres), at RPMs up to 7,500 and boost pressures up to 30 psi. The literature does not mention any trouble with the centre bearing.

[It is hoped that any visitor to this website who has detailed knowledge of this remarkable bearing will be good enough to contact the author by the E-mail link "enquiries@grandprixengines.co.uk" to improve the description. It is also recognised that its conception *may* be due to Percy Riley.]

Note 129

Mercedes F1 Thermal Efficiency and Power output

Data comes from various Mercedes press releases as reported on the i/net.

Maximum Fuel Flow Rate

The current FIA F1 regs from 2014 still limit the fuel flow rate on a rising scale up to a max at 10,500 RPM and above at 100 kg/hr.

[The race ration was 100 kg but was raised to 105 in 2017 when the wider cars and tyres were introduced.]

This is with the regs fuel of 94.25% petrol + 5.75% bio.

2018 TV showed all engines ran up to 12,000 RPM, 14.3% above the max fuel speed. It may be that the mixture is set at ,say, 8% rich at 10,500 and progressively weakens to 5% lean at 12,000, which should be tolerable without burning out pistons or exhaust valves.

Ideal Power Output

Mercedes have stated that the reg max fuel flow rate equates to Ideal Power of 1,240 kW. The calorific value of their Petronas fuel mixture is therefore 12.4 kW.Hr/kg.

[Equivalent to 1547.72 x 12.4 = 19,192 BTU/lb. This is 2.5% higher than the 18,718 BTU/lb which was calculated for generic petrol + ethanol in a synthesis of 24 June 2014.]

1,240 kW = 1,662 BHP = 1,685 PS.

As the Mercedes Power Unit from Brixworth is built in a German-owned firm it is probable that their powers are quoted in metric HP, i.e. Pferdstarke, PS.

2014 Thermal Efficiency (ThE) and Peak Power (PP)

A 2015 release stated that ThE was 45% in 2014,
so PP = 0.45 x 1,685 PS
= 758 PS.

This power includes the power from the current taken from the battery, which of course is derived originally from the fuel. By FIA reg the additive power is limited to
120 kW = 163 PS;

Therefore the Internal Combustion Engine (ICE) had a power of:- 758 – 163 = 595 PS.

This agrees with the usual mention in the Press at the time of 600 “Horsepower”.

2018 ThE and PP

A Sept. 2017 release stated that ThE was then “more than 50%”.
Try 51.5%,
then PP = 0.515 x 1,685 PS
= 868 PS

Deducting the max permitted power of 163 PS from the Energy Recovery System, as before, gives ICE power as:-
= 705 PS

This is 110 PS greater than 2014, where the Mercedes release quoted + 109 PS.

It was stated that the figure was not as run in 2017, so it is assumed it was for the spec. to be raced in 2018.

Possible sources of extra ICE Power

Mercedes did not give any data on how the approx. 18% extra ICE power was produced in 4 years of FIA-controlled development, but they might include:-

- Higher Combustion Efficiency from “Turbulent Jet Ignition”;
- Higher Compression Ratio;
- Higher TurboCharger Efficiency .

Note 130

Ernest Henri and Louis Coatalen, 1912 - 1922

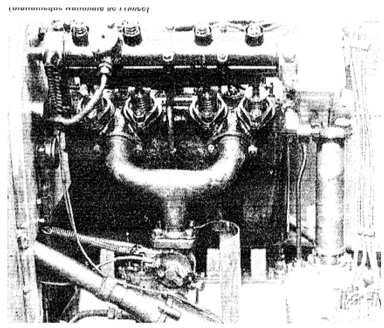
In racing engines, during the period 1912 -1922, Ernest Henri and Louis Coatalen first competed against each other, then Coatalen copied him and finally he employed Henri. This note describes the relationship.

On PP. 4 /5 are shown the engines designed by Henri, first for Peugeot 1912 – 1914, then for Ballot 1919 – 1921 and finally for Sunbeam in 1922.

1912 -1914.

In 1912 Louis Coatalen, Managing Director and Chief Engineer of Sunbeam, entered a team of basically tuned-up side-valve touring cars as "*Voitures Legere*" in the Coupe de L'Auto (C de L'A) to be run concurrently with the Grand Prix de l'ACF (FGP) at Dieppe. The new Peugeot team in the big *Formule Libre* race had been designed by Henri with DOHC and 4 valves per cylinder, and he also produced the design of a new 3 litre for the smaller event (badged as "LIon" to continue the brand of their previous voiturettes). Until publication of DASO 1218 (see References below) it was assumed that this smaller engine was also DOHC and 4 v/c. A photo below shows that it was DOHC but only 2 v/c.

The Sunbeams thrashed the singleton Peugeot, taking 1st,2nd and 3rd places. The Lion DNF with engine trouble. It seems likely that it had suffered inadequate preparation in the effort to finish the Grand Prix cars. The power of the Lion is not known but, as the Sunbeams had 74 HP; presumably it had less.



34 Automobile August 2012

This Sunbeam success encouraged Coatalen in 1913 to enter both the C de L'A and the FGP, still using side-valve engines. The former were similar 4-cylinders to 1912 with 17% more power and the latter 6-cylinder to the same layout. The Peugeots for both races were completely re-designed and improved by Henri, both now DOHC and 4 v/c.

This time it was the Sunbeams which were beaten, the big cars in the June GP coming 3rd (2.5% slower than the winner) and 6th with 2 x DNF; the smaller in September also 3rd (3% slower than the winner) and 2 x DNF.

Although Coatalen had designed overhead-valve racing engines in 1910 and 1911, he clearly now concluded that he needed much more modern technology. He took the simplest way to that. The history is detailed in DASO 1086, how Sunbeam "reverse-engineered" a 1913 3 Litre Peugeot hi-jacked from an exhibition tour. Laurence Pomeroy has told the anecdote that, when his father teased Coatalen about this "Chinese copy "which was produced by Sunbeam for the 1914 RAC Tourist Trophy, the Breton replied "*Ah, Pomeroy, it 'eez a vize man who copies vizout altair*"! Actually, the bore was increased by 3.5 mm, to take advantage of the peculiar TT regulation 3,310 cc limit. Also, although the external differences were concealed by sheet-metal covers at the time (DASO 24 at P.54), one engine was fitted with finger cam followers in place of the rather complicated Henri system for taking cam side thrust (See 1st Naturally-Aspirated Era (1NA) at PP.7 & 8). A team of 4.5 Litre GP cars was also built to the same pattern as the "Henri" TT engine, but to a 7.3% lower S/B than Henri's 1914 engine and using the finger followers – not quite "*vizout altair*"!

Results were not what Coatalen would have hoped. Certainly the RAC race in June was won, 20 minutes in front of the 2nd place Minerva, but 2 cars DNF (an experimental big-end failure and a seized prop. shaft joint). In the July GP only a 5th place was obtained, 4.5% slower than the winning Mercedes, and 2 cars DNF with a big-end and a piston failure.

In 1914 the C de l'A to be run in September was limited to 2.5 Litre engines. Henri followed the scaling-down principle to design his entry. Coatalen used the much-simpler and cheaper method of fitting a short-stroke crank into his TT engine, it being 117 mm in place of 156. The works test curve for this was given in DASO 24; it was rated at 4,000 RPM to produce 92 HP (below the peak of the curve), the same as the bigger engine rated at 2,800 RPM. The MPS for these engines was 15.6 m/s and 14.6, respectively. The value of BThE was 28% and the EV was 79%. Insofar as a comparison is valid between a genuine test and a report, the Henri engine was 13% less powerful, at 80 HP (DASO 5)*. The Sunbeam would have been heavier, unless the cast-iron cylinder block was shortened (not known). The rivals were never put to the acid test of competition, because the war begun by Germany with Russia on 1st August 1914 caused the race to be cancelled.

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*DASO 1218 quotes a test figure for the 2.5 litre Peugeot by the official "*Services des Mines* "as only 71 CV @ 2,920 RPM. On the other hand their figure for the x2 4.9 Litre Ballot was 150 CV @ 2,900 RPM. Pity the motoring historian!

<u>1919 – 1921</u>

After the war Ernest Henri was engaged by Edouard Ballot to design a car for the1919 Indianapolis 500 miles race, to fill a contract which the 1914 winner, René Thomas, already had to compete. It seems that Henri had already laid out for Thomas a novel 8-cylinder engine based on two of his 1914 2.5 Litre engines, with the bore reduced by 1 mm to meet the then-Indy-limit of 300 cubic inches (4,916 cc) (DASO 1218). Other novelties were:- inverted cup tappets; and enclosure of the valve springs (therefore oil-cooled). The 4 cars were completed in record time, proved to be the fastest at the Speedway, but only secured 4th and 10th places because American wheels, made at the last-minute, failed and caused 2 crashes and enforced speed reductions and pit checks on the survivors.

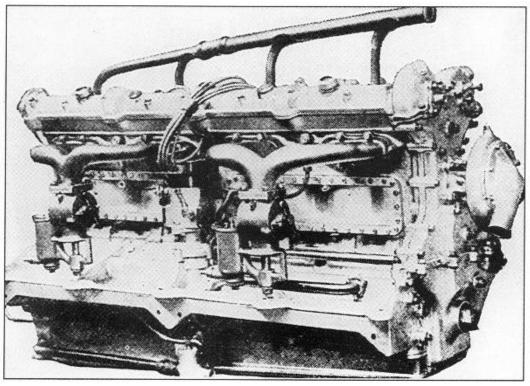
Undaunted, Ballot competed at Indy again in 1920, with engines scaled-down by Henri to the new limit of 183 cid (2,999 cc) (which would also be adopted as 3 Litres for the 1921 FGP revival). Again fastest, again failing to win when the Ballot leading easily had a partly blocked fuel pipe, so only 2nd and 5th. When raced in the 1921 GP another 2nd place was taken, plus a 7th and a DNF (holed fuel tank). A listed Ballot 3rd place was by a remarkable 2 Litre sports car, also designed by Henri

Meanwhile, Sunbeam in 1920 had been amalgamated with Clement-Talbot and Darracq to form the STD group. Coatalen retained his previous posts. The group decided to enter the 1921 GP. It cannot be entirely a coincidence that the engine was 8 cylinders of exactly the same B x S as the 1920 Ballot, or that it had the new inverted cup tappets, which Henri had tried to keep secret in 1919. However, entirely new for an automobile engine was the Al-alloy combined block + head with open-ended steel shrunk-in dry liners, now trusting the alloy to withstand combustion temperature and pressure**. Marc Birkigt had introduced Al-alloy static structure in his 1914 V8 aero engine, but could not let the alloy of the time take this duty, so had used closed-end liners. Coatalen's design therefore pre-dated

the 1925 Rolls-Royce "F" aero engine (wet liners) and the 1932 Alfa Romeo type B GP engine (dry liners) in its reliance on a light alloy head. The war-time research into the material had no doubt permitted this novelty. The Sunbeams should have been lighter than the Ballots, which had cast-iron block + head. Also, their plain crank bearings would have been lighter than the Ballot ball bearings.

The 3 litre STD cars were not ready for the 1921 GP, the entries of all 7 were cancelled, then resurrected at the last moment for 4 cars. They suffered tyre trouble and 5th was the best result. The 3 Litre did redeem itself by winning the 1922 RAC Tourist Trophy. An illustration of the type is given below (RHS. LHS, 1919 Ballot No. 1003).

Photo courtesy of Eddie Berrisford



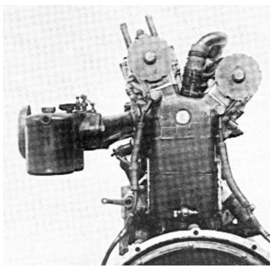
DASO 24

That the basic design was not at fault was shown by the racing history of the 4 cylinder 1.5 litre voiturettes derived from the eight. They became known as the "*invincible Talbot-Darracqs*" (badged as such), since over 1921 and 1922 they won all 11 events in which they were entered.

**That the liners <u>were</u> open-ended is shown by the description in DASO 24:- "*The two lightweight aluminium alloy cylinder blocks had shrunk-in steel liners and <u>screwed-in phosphor-bronze valve seats</u>*"(author's underlining). As three of the 3 Litre cars still exist there must be many people who could confirm or correct this statement. The author hopes one will contact him via "Enquiries".

<u>1922</u>

Coatalen apparently was not satisfied with the 1921 3 Litre and *finally* engaged Henri directly to do something better for the 1922 GP, which was limited to 2 Litres. The success of the Talbot 4 cylinder 1.5 Litre perhaps encouraged Henri to return to that number of cylinders. It was probably unknown that FIAT would use 6. He introduced a "lopsided" 4 v/c disposition, 20^0 offset for the inlets to straighten out the inflow, 40^0 for the exhausts. A 2nd spark plug was inserted under the inlets (see figure RHS from DASO 24).



It was to no avail – the FIATS were faster and all 3 Sunbeams retired with broken valve stems.

This was the end of Ernest Henri as a front-line racing engine designer, after 7 racing years.

Coatalen then once again looked for foreign inspiration. He hired Walter Becchia and Vincenzo Bertarione from FIAT to more-or-less copy the 1922 FIAT for the 1923 GP – which it won, of course. They also redesigned the 1.5 Litre voiturette as a 4 cylinder version of the GP car – and it resumed the winning ways of its predecessor.

PP. 4 & 5 give basic engine data and specific performance analysis

THE ENGINES OF ERNEST HENRI and their copies				
1912	**1913**	**1914**	**1916**	
As designer for **PEUGEOT**				
IL4 7.6 L Grand Prix	**IL4 5.6 L Grand Prix**	**IL4 4.5 L Grand Prix**	The 1912 engine had 5 plain crank bearings, and shaft cam drive. The later engines had 3 ball bearings on a built-up crank, in a barrel crankcase, and spur-gear cam drive..	
S = 200 mm/B = 110 mm	S = 180 mm/B = 100 mm	S = 169 mm/B =92 mm		
7,603 cc S/B = 1.818	5,655 cc S/B = 1.8	4,494 cc S/B = 1.837		
DOHC 4 v/c	DOHC 4 v/c	DOHC 4 v/c		
140 HP @ 2200 RPM	115 HP @ 2500 RPM	112 HP @ 2800 RPM		
IL4 3 L Voiturette	**IL4 3 L Voiturette**	**IL4 2.5 L Voiturette**		
S = 156 mm/B = 78 mm	S = 156 mm/B = 78 mm	S =140 mm/B =75 mm		
2,982 cc S/B = 2	2,982 cc S/B = 2	2,474 cc S/B = 1.867		
DOHC 2 v/c NB!	DOHC 4 v/c	DOHC 4 v/c		
	90 HP @ 2900 RPM	80 HP @ 3000 RPM		
SUNBEAM	Copied from the 1913 Peugeot Voiturette plus 3.5 mm on B towards reaching TT 3,310 cc limit. One engine had Finger Cam Followers.	**IL4 3.3 L Tourist Trophy**	**IL6 4.9 L Indianapolis**	Not known how Coatelen got permission to build and race a new car at Indianapolis in neutral USA in wartime.
		S = 156 mm/B = 81.5 mm	S = 156 mm/B = 81.5 mm	
		3,255 cc S/B = 1.914	4,883 cc S/B = 1.914	
		DOHC 4 v/c	DOHC 4 v/c	
		92.5 HP @ 2800 RPM	156 HP @ 3000 RPM	
	Modified from Peugeot basis in S/B and in having Finger Cam Followers.	**IL4 4.5 L Grand Prix**	To suit Indianapolis 300 cu. inch (4,916 cc) rule	
		S = 160 mm/B = 94 mm		
		4,442 cc S/B = 1.702		
		DOHC 4 v/c	Al-alloy pistons, as advised to Louis Coatelen by Walter Bentley. The 6-cylinder had bad crank vibration at 3,000 RPM.	
		108 HP @ 2800 RPM		
	As 3.3 L with reduced S.	**IL4 2.5 L Voiturette**		
		S = 117 mm/B = 81.5 mm		
		2,442 cc S/B = 1.436		
		DOHC 4 v/c		
		92 HP @ 4000 RPM		
Sources for Power figures:- DASOs:- 4,5,24,26.				

1919	1920	1921	1922
As designer for **BALLOT**			**FIAT**
IL8 4.9 L Indianapolis	**IL8 3 L Indianapolis**	**IL8 3 L Grand Prix**	**IL6 2 L Grand Prix**
S = 140 mm/B = 74 mm	S = 112 mm/B = 65 mm	As in 1920 Tndy.	S = 100 mm/B = 65 mm
4,817 cc S/B = 1.892	2,973 cc S/B = 1.723		1,991 cc S/B = 1.538
DOHC 4 v/c	DOHC 4 v/c		DOHC 2 v/c
140 HP @ 3000 RPM	110 HP @ 3800 RPM		112 HP @ 5,000 RPM
1914 Voiturette doubled-up, but B reduced by 1 mm to suit Indianapolis 300 cu. inch rule	Inverted Cup tappets in Ballots	?	Chief Engineer
			Guido Fornaca
	SUNBEAM, continued		As designer for **SUNBEAM**
		May 1921	
		IL8 3 L Grand Prix	
		S = 112 mm/B = 65 mm	
		2,973 cc S/B = 1.723	
	Al-alloy block/head. All plain bearings. 1 carburetter per 2 cylinders. Inverted cup tappets.	DOHC 4 v/c	
		108 HP @ 4000 RPM	**IL4 2 L Grand Prix**
		September 1921	S = 136 mm/B = 68 mm
		IL4 1.5 L Voiturette	1.976 cc S/B = 2
		S = 112 mm/B = 65 mm	DOHC 4 v/c
		1,487 cc S/B = 1.723	88 HP @ 4200 RPM
		DOHC 4 v/c	Suffered valve failures.
		51 HP @ 4000 RPM	
		Badged as Talbot-Darracq	

All engines NA on petrol, which would be rated retrospectively at no better than 50 Octane.

Specific Performance								
Date	1914	1914	1914	1916	1921	1922		1922
Make	Sunbeam	Sunbeam	Sunbeam	Sunbeam	Sunbeam	Sunbeam		FIAT
Type	TT	GP	C de l'A	Indy	GP	GP		GP
P/V - HP/Litre	28.42	24.31	37.67	31.95	36.33	44.53		56.25
BMEP - Bar	9.08	7.77	8.43	9.53	8.13	9.49		10.07
at MPS - m/s	14.56	14.93	15.6	15.6	14.93	19.04		16.67

	Comparative Performance of 1922 Engines			
	Sunbeam		FIAT	FIAT/S'beam
BMEP - Bar	9.49		10.07	x1.061
at MPS - m/s	19.04		16.67	x0.875
100/Smm	0.735		1	x1.36
P/V - HP/Litre	44.53		56.25	x1.263 = 1.061x0.875x1.36

Comparative analysis of 1922 Sunbeam & FIAT

For engines designed on a Volume Specific Power basis the relevant relation is:-

$$P/V \propto [(BMEP) \times (MPS)]/S$$

where, at a given “State of the Art”, BMEP is the attainable “stress” in the working fluid and $(MPS)^2$ represents the acceptable stress in the reciprocating parts, leaving 1/S to determine P/V.

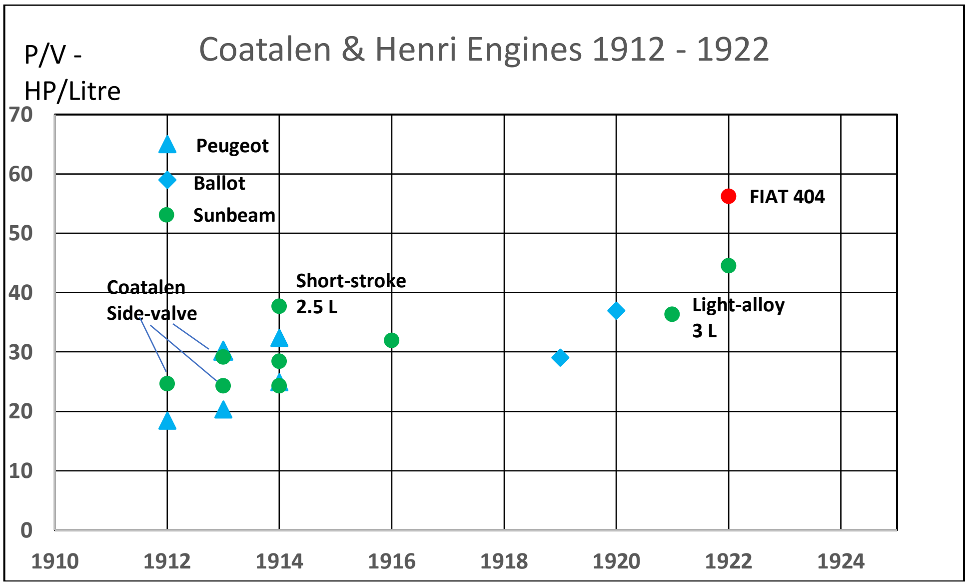
The table above compares the 1922 factors for the Henri-designed Sunbeam with the winning FIAT designed by a team under Guido Fornaca.

BMEP: -FIAT introduced a new 2 v/c hemispherical combustion chamber with (Inlet Valve Area/Piston Area) (IVA/PA) of 0.38, where the Sunbeam had the typical Henri pent-roof chamber with “negative squish” to accommodate over-large 4 v/c with IVA/PA of 0.48. The result was modestly in FIAT’s favour.

MPS:- FIAT made a better choice with the materials of the time for a 500 mile GP. The Henri figure of 19 m/s contrasted with the value of 14 which he had selected for his 500 mile Indy engines. In practice, the engines were much more highly stressed than 19 m/s because it was decided to lower the axle ratio to improve acceleration and, coupled with tachometers reading 300 RPM too low (DASO 24), the engines were run up to nearly 5,000, 22.7 m/s or 42% greater stress. Apparently this was discovered before the race, but fatigue life must have been used up; 2 engines suffered valve failures by only 8% distance, although Segrave’s survived to 42%. DASO 26 reports Henri remarking bitterly after the race that Coatalen had insisted on some modification which had caused the failures, but his high design stress aggravated by over-revving must share the blame.

1/S:- This leaves the principal *performance* advantage of the FIAT (as opposed to its reliability*, excepting* its axle construction – 2 of those failed) as being a 6 cylinder engine with 36% shorter stroke. Fornaca had chosen to reduce his 1921 IL8 3 L to IL6 2 L with B and S adjustments. Henri could have used the same method but built a highly-stressed 4 instead.

An overview of the Coatalen & Henri engines is given on the chart on P.6 below.



More details of the 1913 3L Peugeot are given as Eg. SO5 in Appendix 1 and Significant Other. The 1920 3L Ballot is Eg. SO6 in the same website references.

Conclusions

Louis Coatalen was a firm believer that "*Racing improves the breed*" by squeezing out complacency from design. He certainly spent the money to put his belief into practice, although it is not known whether this raised sales of production cars. Between 1912 and 1922 Sunbeam racing was in continual relation to Ernest Henri, firstly in competition, then in copying and finally in employment.

In the international sphere, Sunbeam gained a stunning success in the 1912 Coupe de l'Auto (C de l'A) with their side-valve cars, won the 1914 RAC Tourist Trophy with a Peugeot copy and again in 1922 with an engine having Ballot resemblance, competed four times in the Grand Prix de l'ACF (FGP) but did not win it..

Although always prepared to copy, Coatalen *did* introduce innovations:- a bold short-stroke engine in 1914 for a 2.5 litre C de l'A which it might well have won if the German war had not intervened; and a light-alloy engine for the 1921 FGP which was hampered both by inadequate preparation and poor tyres, but at least provided the basis for the "*invincible Talbot-Darracq*" voiturettes of 1921 - 1922.

It awaited the near-copy of the 1922 FIAT before the prize of the FGP was won at the fifth attempt.

Afterthoughts

1. In choosing a shorter-stroke to produce the 1914 2.5 L Voiturette engine, the valve gear had to run at 4,000 RPM instead of 3,000. Nothing is known about how it behaved. The change to finger cam followers probably helped.
2. It will be seen from the Overview chart that Sunbeam 2 v/c side-valve engines had higher P/V than Peugeot DOHC 4 v/c, according to the available data. If so, it is unlikely that the Sunbeam figures were sustainable, because of exhaust valve burning. This subject is discussed further in Note 25.

References

DASO 4. Grand Prix Car Revised Edition Vol. 1 L. Pomeroy MRP 1954.
DASO 5 Profile No. 73 W Court 1967.
DASO 24 Sunbeam Racing Cars 1910-1930 A. Heal Haynes 1989.
DASO 26 Classic Twin Cam Engine G. Borgeson..Dalton Watson 1981.
DASO 1086 Sunbeam Aero Engines A. Brew Airlife 1998.
DASO 1218 *Automobile* August 2012 Article by S. Faures Advised by courtesy of Keith Eames.

This Note is dedicated to the memory of the *White Mouse* stable, whose exploits over 1935 to 1946 as described by Prince Chula of Siam in his books *Wheels at Speed, Road Racing 1936, Road Star Hat-trick* and *Blue and Yellow* provided so much pleasure to this author.

1.5 Litre ERA Lap Speeds, 1935 – 1968

The 1.5 Litre ERA offers a unique way of examining Lap Speed (LS) across a variety of racing circuits because:-

- It has had such a long career – 1935 to date (taken up to 1968 here);
- Power and Weight are well-known (within the limits that any racing car data is known!), provided allowance is made for whether the engine was Pressure-Charged by Jamieson Roots supercharger (177 BHP) or Zoller vane-type (230 BHP);
- The range of circuits for which LS is known is very wide – from Monaco (56.8 MPH) to Brooklands Outer (124.8).

Clearly there was a range of abilities in producing LS and also learning, as a driver became accustomed to a car over the years, so some scatter is unavoidable. The drivers are listed on P.2

The LS (MPH) by driver and circuit are given on P.4 , together with the Track Factor (TF). TF was defined in "Progress over 64 years of Grand Prix racing, 1951 to 2014" as:-

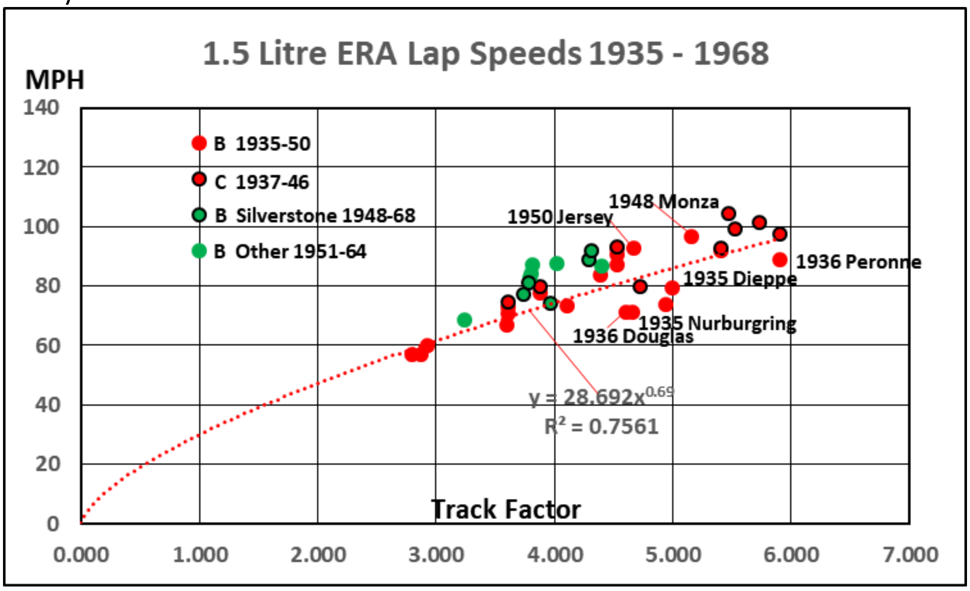
$$TF \propto (1609\,x\ \ L)^{0.4})/T^{0.3}$$

where L = Lap length in miles (1609 converts to metres to be consistent with "64 Years Progress");

T = Total turning per lap, degrees.

The temperature term in the ref. is discarded.

The best analysis of the data is charted below.



The trendline shown was based on B-type cars over 1935 to 1950. It is a poor correlation, but does provide a picture of the data. Generally-faster C-type figures were added separately, as shown. In 1937 these had Porsche-type independent front suspension (IFS).

What is surprising is that LS after 1948 on English circuits were generally so much faster at a given TF. The initial thought was that this was due to grippier tyres, but the Dunlop R1 in use from 1946 to 1958 was, it is believed, essentially the same as the pre-War "5 Stud" type. It was not until 1958 that the better R5 became available and was certainly used on some ERAs, and this may account for some higher speeds later.

This author suggests that much of the increase was due to the circuits being much safer than pre-War. All circuits up to1939 and some after were extremely dangerous. A small deviation could cause a fatal accident with the car wrapped round a tree, as happened to Dick Seaman at Spa in 1939. Even the boldest drivers had to temper valour with discretion. Post-war British circuits were usually adapted from airfields, with wide run-ways and space for off-track excursions, and drivers expected the same consideration with later purpose-built tracks.

To complete the range of circuits, figures are given on P.4 of two super-fast cases:-

- Charles Martin's lap of the Berlin AVUS at 122.7 MPH in 1937. This was after the North Curve had been rebuilt with a 43 degree bank. The rest of the lap was two long straights joined by the slightly-banked South curve.
- St John Horsfall's lap of the Brooklands Outer circuit at 124.8 MPH in 1939. The circuit is described in Brooklands Outer Circuit Lap Speeds. This exploit must have been at max. RPM with a road-racing axle ratio, because, geared for the track, it should have been at least 10 MPH faster.

1.5 Litre ERA Lap Speeds:- 1935 - 1968

Key to Tables

B	1 Jamieson Roots-type Supercharger (100 or 120 mm rotor length).
	Details are given in Appendix 5 at col. X, and a section in Illustrations for App.5 at P.25.
	There is a good B-type cutaway drawing in Note 128.
C	1 Zoller Vane-type Supercharger.
	Details are given in Appendix 5 at col. AB. See also Illustrations for App.5 at P. 25.

Drivers

M	Raymond Mays
B	"B. Bira"
S	Dick Seaman
L	Marcel Lehoux
H	Lord Howe
F	Pat Fairfield
CM	Charles Martin
R	Tony Rolt
D	Arthur Dobson
G	Bob Gerard
ST	Brian Shaw-Taylor
BM	Bill Morris
W	Ken Wharton
GW	Graham Whitehead
CH	Cuthbert Harrison
JH	St John Horsfall
PW	Peter Walker
WM	Bill Moss
JC	Jimmy Clark

B Bira, R2B Romulus at Crystal Palace in 1938.
Photo by Louis Klementaski

The original caption reads:- *"May it serve as a reminder that the 4-wheel drift is not a post-War invention".*

R2B probably cost £1,050 in July 1935 (DASO 449. £75,000 in 2019 money). It was fitted originally with a 100 mm supercharger (L. Snellman) driven at 1.8 x crank RPM (DASO 449). For the 1936 event on the fast Peronne circuit the 120 mm unit from R5B was fitted (L. Snellman), probably with the same driving gears to increase the boost, although a 1.4 ratio was specified for the bigger blower. Bira won the race. De Ram spring dampers were fitted by the Chula équipe after 1936, as seen, replacing the original Hartford simple friction-type. These were true "Shock absorbers" as they were also friction-type but the plate pressure was varied hydraulically proportionately to the speed of the operating arm. The de Ram type had been fitted on the 1933 Bugatti T59 and were said to improve greatly the ride over bumpy roads. They cost £200 per set, fitted (*M. Sport* April 1935. 19% of the original car cost!). Also fitted later with Lockheed 2 leading shoe hydraulic brakes, replacing original Girling mechanical.

R5B Remus ex-Chula, which was driven by Bill Moss, 1956-1959. Ludovic Lindsay aboard here.
Photo credit:- Goodwood

R5B probably cost £1,800 in 1936 (*Road Racing 1936.* £129,000 in 2019 money) because ERA raised their prices after the car showed itself to be successful. It was supplied with a 120 mm supercharger driven at 1.4 x crank RPM (L.Snellman & DASO 449). This may have been more efficient than the higher RPM 100 mm blower. As reported above this unit was removed and fitted to R2B in 1936. Current spec. not known.

1.5 Litre ERA Lap Speeds:- 1935 - 1968

Year	Type	Driver	Circuit	Code	L - Miles	T - Degrees	Practice - P or Race - R	TF	Lap Speed - MPH for B-type	Lap Speed - MPH for C-type
1935	B	M	Nurburgring	NU	14.173	3840	R	4.655	71.3	
1935	B	M	Dieppe	DE	5.005	760	R	4.990	79.3	
1935	B	S	Berne	BE	4.524	920	R	4.526	87	
1936	B	H	Monaco	MO	1.96	1494	P	2.800	56.8	
1936	B	B	Donington	DO	2.551	916	P	3.604	70.6	
1936	B	L	Douglas 36	DS36	4	736	P	4.607	71.2	
1936	A	F	Berne	BE	4.524	920	P	4.526	92.96	
1936	B	B	"	'	4.524	920	R	4.526	90.5	
1936	A	F	Donington	DO	2.551	916	P	3.604	72.9	
1936	A	F	Peronne	PE	6.06	560	P	5.904	88.7	
1936	B	B	Cork	CO	6.088	754		5.410	92.08	
1936	B	B	Limerick	LI	2.76	660	R	4.103	73.34	
1937	B	B	Turin-Valentino	TU	1.818	1242	P	2.872	56.9	
1937	C	M	Donington	DO as '36	2.551	916	P	3.604		74.7
1937	C	M	Albi	AL	5.53	617	R	5.529		99.1
1937	C	M	Phoenix Park	PP	4.261	450	R	5.476		104.36
1938	C	B	Cork	CO	6.088	754	R	5.410		92.86
1938	C	M	Peronne	PE	6.06	560	P	5.904		97.39
1938	B	R	Leinster	LE	5.917	1454	R	4.392	83.53	
1938	C	M	Berne	BE	4.524	920	P	4.526		93.1
1938	B	D	Donington 1937	DO37	3.125	936	P	3.883	77.8	
1939	C	B	Donington 1937	DO37	3.125	936	P	3.883		79.63
1939	B	B	Crystal Palace	CP	2	1332	P	2.922	60.2	
1939	C	B	Rheims	RH	4.861	460	P	5.734		101.6
1946	B	B	Geneva	GE	1.855	604		3.595	66.8	
1946	B/C	B	Ballyclare	BL	4.142	708	R	4.726		79.7
1948	B/C	CH	Monza	MA	3.915	490		5.160	96.5	
1949	B	G	Douglas37	DS37	3.87	558	R	4.940	73.9	
1950	B	G	Jersey	J	3.198	520	P	4.675	92.6	

1948	B	G	Silverstone 1948	SI48	3.67	1080		3.967	74.1	
1949	B	PW	Silverstone 1949	SI49	3	964		3.787	81.1	
1951	B	G	Silverstone 1950	SI50	2.889	604		4.292	88.9	
1952	B/C	GW	Silverstone Club	SC	1.608	440		3.733	77.2	
1968	B	BM	Silverstone	SI52	2.927	604		4.314	91.8	

1951	B	ST	Goodwood	GO	2.4	586		4.021	87.7	
1952	B	KW	Castle Combe	CC	1.84	494		3.806	84.1	
1953	B/C	GW	Crystal Palace	CP53	1.39	580		3.242	68.55	
1953	B/C	GW	Goodwood 1952	GO52	2.4	700		3.812	87.3	
1961?	B	WM	Oulton Park	OP	2.761	860	R	3.791	82.97	
1964	B	JC	Rouen	RO	4.065	874		4.403	86.75	

Banked Circuits

1937	B	CM	AVUS	AV	11.98	446	R	8.302	122.7	
1939	B	JH	Brooklands Outer	BO	2.767	414		4.724	124.82	

Any Corrections & Additions

Hundreds of people have operated ERAs over the years, and the author would be pleased to receive from them any corrections or additions to this Note 131 via the Enquiries Contact.

Conclusions

- Modifying Total Turning (T) to discard slight curves and magnify sharp corners (very frequent on pre-War circuits) might improve the correlation. This would need a defined method, to avoid judgement. A fresh Multi-Variable Regression Analysis might also be helpful, considering the very-different characteristics of an ERA vs. 2013 cars.
- The advantage of the C-type IFS on bumpy pre-War circuits cannot be separated from the higher power and cannot be shown geometrically.
- The suggested "mental" gain from safer circuits also cannot be shown geometrically.

Comment
Although the original intention of this note was not achieved, re-visiting ERA history has been a pleasure and the author hopes that visitors will also enjoy it

References
DASO 449 Racing an Historic Car P.Hull MRP 1960.
www.kolumbus.f1/leif.snellman The Golden Era of GP racing, 1926-1940.

Note 131 described the longevity of the B-type ERA racing career as an opportunity to use its performance over a wide variety of circuits as a way of correlating Lap Speeds (LS, MPH) with their geometry. From previous Multi-variable Regression Analysis (MRA) work on the 2013 Grand Prix season the relation used was:-

LS proportional to $(1609 \times L)^{0.4} / (T)^{0.3}$
Where 1609 converts L (Lap length) in miles to metres;
and T = Total degrees turning in the lap, Left + Right.

The RHS of the relation was named Track Factor, TF.

This check produced the rather poor result:-

$LS = 28.7. (TF)^{0.69}$
with the co-efficient of correlation of $R^2 = 0.76$.

This could have been re-cast as:-

$LS = 28.7.(1609 \times L)^{0.276} / (T)^{0.207}$

Factor of difficulty

The author felt that visitors were entitled to some further work on the subject of relating car performance to circuit characteristics.

Note 131 concluded that, while the angles comprising T were not doubtful, there needed to be some "Factor of Difficulty" applied to their values to improve the correlation with LS. To do that required some method laid down to minimise subjective judgement. Given below is the method used next, admitting that the angle multiplications were purely arbitrary to see whether the result seemed sensible.

Method for determining Modified Turning Angle (TM)

Given the basic details measured for total turning (T), with Left and Right angles identified and with the difference checked as 360^0 , the Modified Turning Angles are determined as follows, by multiplying each angle by a Factor of Difficulty (FOD).

. Measured Turning Angle — FOD multiplication

- **40^0 or less** — Measured angle **x 0** — Treated as "Non-friction-limited", by the car path radiused within the road width.
- **Between 80^0 and 100^0**
  Sharp corner meaning that the car path is radiused only in road width:- Measured angle **x 2**
  Radiused corner — Take angle **as measured**
- **Beteen 150^0 and 180^0**
  Sharp Hairpin (as above) — Measured angle **x 3**
  Radiused — Take angle **as measured**
- **Chicane** — Measured angles **x 1.5**

The data for the B-type ERA with the values of TM is given on P.2. A MRA was made with input of L and TM and after some trial and error a new Track Factor TF4 was adopted. This was:-

$TF4 = (1609 \times L)^{0.22} / (TM)^{0.26}$.

B-type ERA

Year	Driver	Circuit	L Miles	T^0	TM^0	LS-MPH	TF4
1935	Raymond Mays	Nurburgring	14.17	3840	4176	71.3	1041
1935	Raymond Mays	Dieppe	5.005	760	1111	79.3	1.168
1936	Lord Howe	Monaco	1.958	1494	2028	56.8	0.812
1936	Marcel Lehoux	Douglas	4.000	826	1097	71.2	1.115
1936	Pat Fairfield	Donington	2.551	916	1013	72.9	1.031
1936	B. Bira	Albi	5.531	617	893	95.3	1.264
1936	Pat Fairfield	Peronne	6.06	560	922	88.7	1.279
1936	Pat Fairfield	Berne	4.524	920	654	90.5*	1.311
1936	B. Bira	Cork	6.088	754	1075	92.08	1.230
1936	B. Bira	Limerick	2.76	840	1177**	73.34	1.080
1937	B. Bira	Turin	1.818	1242	2335	56.9	0.771
1938	Tony Rolt	Leinster	5.917	1454	1550	83.53	1.111
1938	Arthur Dobson	Donington '37	3.125	936	1012	77.8	1.079
1939	B. Bira	Crystal Palace	2.000	1332	1540	60.2	0.877
1946	B. Bira	Geneva	1.855	604	1095	66.8	0.942
1948	Cuthbert Harrison	Monza	3.915	490	477	96.5	1.379
1949	Bob Gerard	Douglas '37	3.87	826	881	73.9	1.172
1950	Bob Gerard	Jersey	3.199	520	650	92.6	1.217

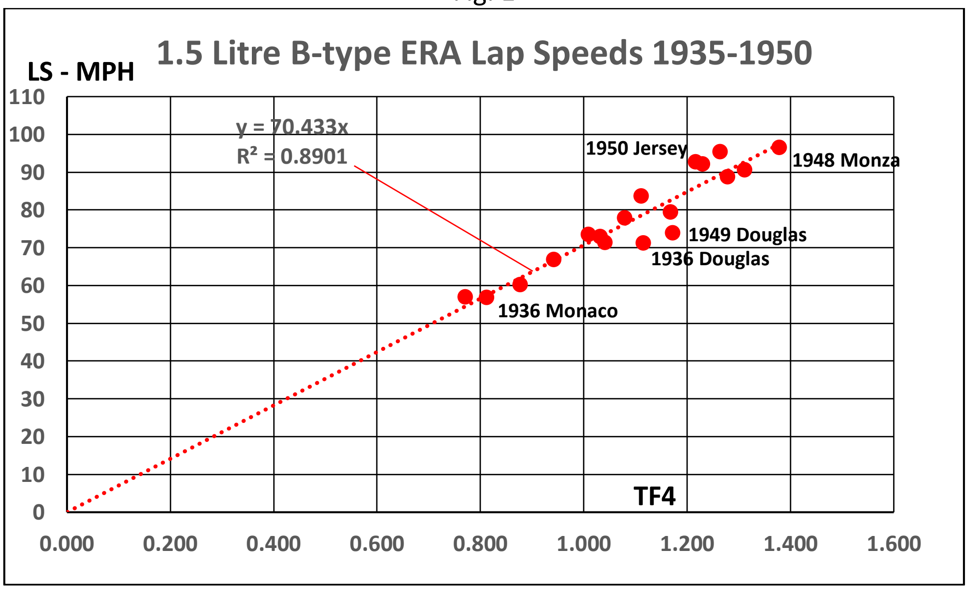
Only the highest B-type values of LS for each circuit are taken here.

*Correcting LS from 92.96 in Note 131 from DASO 1219 (see Refs. below) to the figure from DASO 1221. An advance of nearly 6 MPH over Dick Seaman's 1935 lap was just not possible. There must have been a timing error.

**Including 2 chicanes not shown on the available plan but described in reports and seen on a U-tube of a contemporary newsreel.

Fig. 1 gives the result of plotting LS versus TF4.

Fig. 1



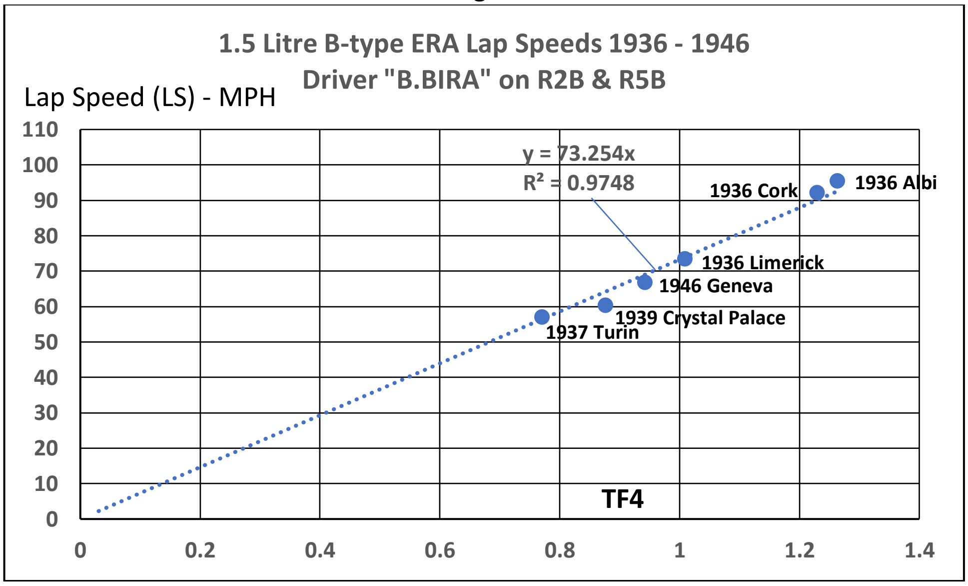
Clearly, Fig.1 is a large improvement on Note 131, since the co-efficient of correlation has increased from 0.76 to 0.89. A result which aligns 1936 Monaco (56.8 MPH) with 1948 Monza (96.5 MPH) cannot be too bad! The scatter is still too high, although the poor result for Douglas 1936 is easily explained: DASO 1221 describes the back stretch of that circuit as follows: " *very difficult abounded with sharp and blind corners high banks and hedges on both sides* ". The just-as-poor result for Douglas 1949 is curious. The circuit, first chosen in 1937, was described then by *The Motor* as a "100% improvement" (over 1936), with a flat-out dash along the nearly mile long promenade, but DASO 1222 says that there followed "*a most difficult climbing section with plenty of zigzag and hairpin bends* " which must explain the low speed. These points are made to illustrate the problem of only being able to use published plans. The high Jersey speed can only be put down to the width of the roads used, easing the apparent two hairpins.

At any rate, the arbitrary FOD multiplications do not seem to be wildly out.

<u>The correlation for individual drivers</u>

Fig. 1 includes 18 laps by 9 drivers. To eliminate scatter from this, the career of "B. Bira" has been examined separately, since this very-talented driver with a well-managed team accomplished 6 of these laps. The result is shown on Fig. 2.

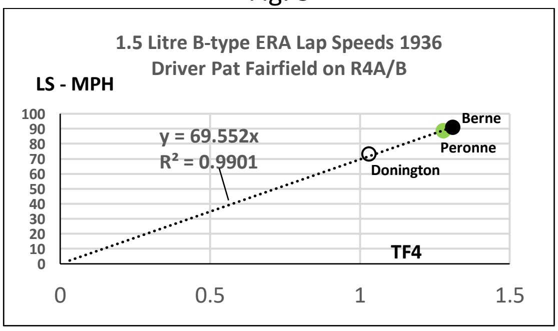
Fig. 2



For this single driver over a wide range of lap speeds, the correlation co-efficient has reached the remarkable value of 0.975.

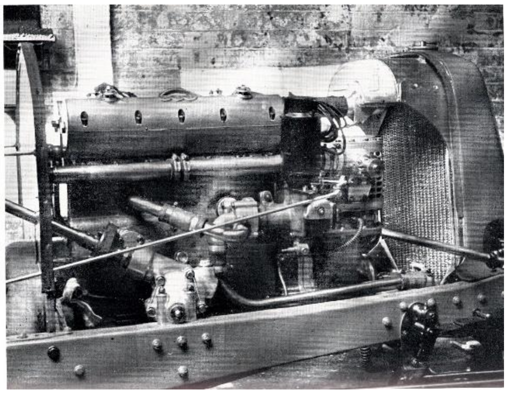
Pat Fairfield in 1936 produced 3 of the laps tabled. His individual performance is even better correlated by TF4, as shown on Fig. 3*

Fig. 3

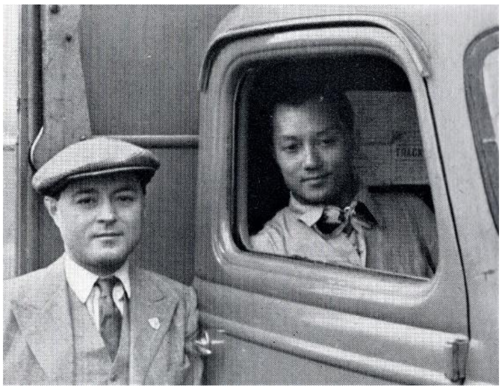


[*The author has chosen marker colours appropriate to the drivers in Figs 2 & 3: blue for Bira, of course; white for Fairfield as an independent at Donington, green for his incorporation into the works team for Peronne and black for Berne when the works decided that green was unlucky. His R4A/B may not actually have been re-painted in team colours.]

Engine of ERA R2B "Romulus".



DASO 1223



DASO 1223

The White Mouse équipe, 1935 - 1948.
Manager Prince Chula of Siam and his cousin and driver "B. Bira". As an honorary member of the British Racing Drivers' Club Bira was eligible to compete for their Gold Star. He won it in 1936,1937 and 1938, ahead of Dick Seaman, Raymond Mays and Arthur Dobson, respectively.

Pat Fairfield
1907 - 1937

PA Images

Conclusion

After a revised MRA specific to the B-type ERA, adopting for the circuit curves the Factors of Difficulty tabled above and looking at the performance of individual drivers, a good correlation has been obtained for lap speed versus modified geometry.

References

DASO 1219 www.kolumbus.f1/leif.snellman The Golden Era of GP racing, 1926 – 1940.
DASO 1221 Road Racing 1936 Prince Chula of Siam..Foulis 1972 ed.
DASO 1223 Blue and Yellow Prince Chula of Siam Foulis 1947.

<u>Open-wheeled Racing and Racing-Sports cars</u>

“PASSION FOR SPEED” (DASO 1229 N. Mason & M. Hales Carlton 2010) provides descriptions and performance tests for 24 cars owned by Nick Mason and driven by Mark Hales. To put the examples comprising open-wheeled racing and sports-racing cars in perspective the author provides here a simple analysis for this 10 car sample, the data being given below.

VMAX = Maximum Speed

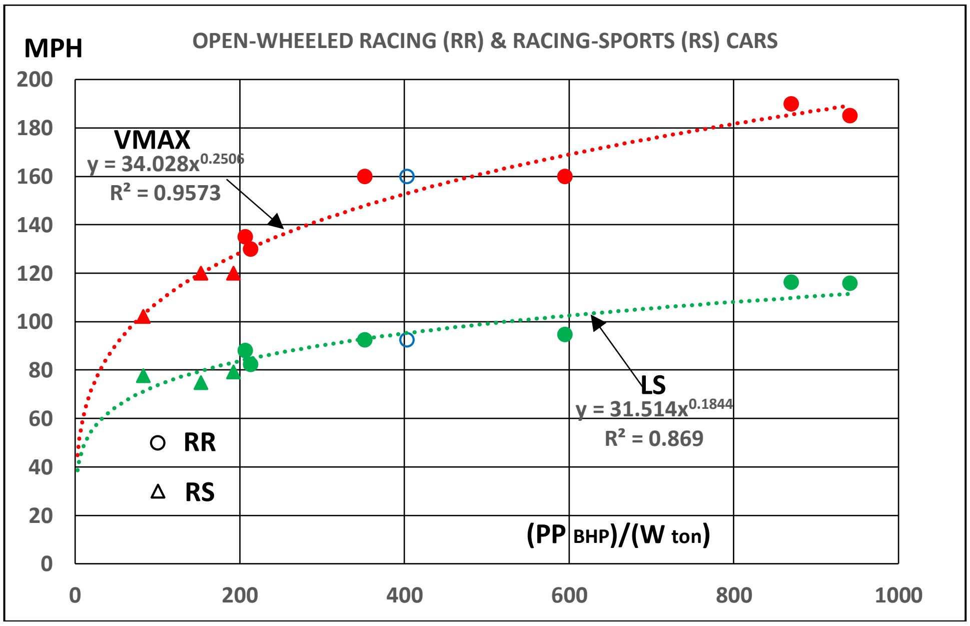
LS = Lap Speed round Silverstone’s pre-2010 South Circuit (2 miles lap assumed)

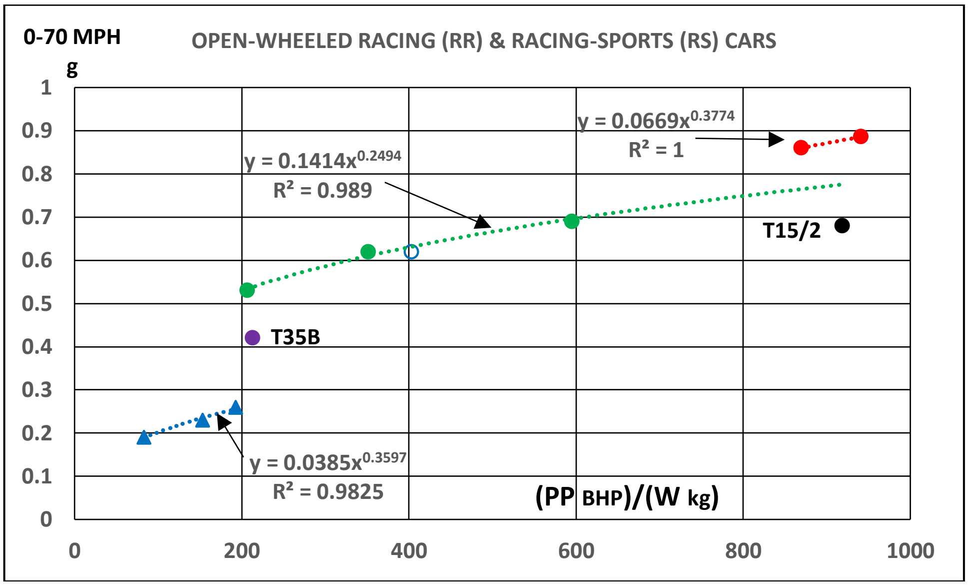
Year	Make	Model		PP-BHP	W* -kg	PP/W BHP/Ton	VMAX-MPH	LS-MPH	0-70 MPH g
1927	Bugatti	35B	RR	160	762	213.3	130	82.4	0.42
1931	Alfa Romeo	8C/2300	RS	155	1027	153.3	120	74.6	0.23
1935	Aston Martin	Ulster	RS	85	1040	83	102	77.6	0.19
1936	ERA	B	RR	150	738	206.5	135	88	0.53
1949	Frazer nash	L M Replica	RS	140	738	192.7	120	79.1	0.26
1953	BRM	T15 Mk2	RR	575	636	918.8			0.68
1957	Maserati	250F	RR	218**	630	351.6	160	92.5	0.62
1961	Lotus	18	RR	250	427	594.9	160	94.6	0.69
1978	Ferrari	312T3	RR	510	596	869.4	190	116.3	0.86
1983	Tyrrell	11	RR	500	540	940.8	185	115.8	0.886

VMAX, LS, and 0-70 MPH are shown versus PP/W on the charts below (0-70 on P.2).

*As in DASO 1229, presumed empty without allowing for driver or test equipment.

**Not known if this is a bench-test power on the car owned – it seems low. Blue circles on the charts show the effect if the power was 250 BHP.





These charts have to be considered as p*ictures,* just to put the cars in perspective, because each line includes some *known* but u*n-quantifiable* variables:_

- VMAX for RS cars includes the losses of silencers and the drag of mudguards and lights;
- LS over the 57 years of manufacture from 1927 to 1983 includes the time-based changes in materials and construction for brakes; for materials, construction and pressure for tyres; for different suspension systems; plus the use of aero systems post-1967 to increase downforce. Notes on these effects have been given in "Progress over 64 years of Grand Prix racing".
- For the acceleration chart the sort of tyres fitted is of *major* importance and, apart from the section dimensions, nothing is recorded about materials, construction, or tyre pressures. It appears from the test results that the Racing-Sports (RS) cars have different tyres from the pure racing cars. Being rear-wheel-drive and allowing for some weight transfer to the rear, any acceleration over 0.7 g must mean a tyre co-efficient of friction above 1. This will be true of the two top points, which may also have some slight beneficial aero downforce at speeds above 60 MPH.

The wide range of the Nick Mason RR and RS cars driven by Mark Hales is shown by illustrations on P.3.

myautoworld.com

Bugatti T35B driven by William Grover-Williams
to win the 1st Monaco Grand Prix in 1929.

oldracingcars.com

Tyrrell 011 as driven by Michele Alboreto to
win the Detroit Grand Prix in 1983, powered by the Cosworth DFY engine.

The author trusts that there will be no objections to the use of this data in a not-for-profit site intended to assist study.